Me 2

(244/1958)

UDC 534,838,1 : 629,12.07

ACTA POLYTECHNICA SCANDINAVICA

MECHANICAL ENGINEERING SERIES No 2

VIGGO A. KJÆR

Vertical Vibrations in Cargo and Passenger Ships

Danish Contribution No 2

COPENHAGEN 1958

ACTA POLYTECHNICA SCANDINAVICA

. . . a Scandinavian contribution to international engineering sciences

Published by the Scandinavian Council for Applied Research and co-sponsored by

The Danish Academy of Technical Sciences

The Finnish Academy of Technical Sciences and The State Institute for Technical Research

The Norwegian Academy of Technical Sciences and The Royal Norwegian Council for Scientific and Industrial Research

The Royal Swedish Academy of Engineering Sciences, The Swedish Natural Science Research Council, and The State Council for Technical Research

Acta Polytechnica Scandinavica is divided into the following sub-series:

Chemistry including Metallurgy series, Ch
Civil Engineering and Building Construction series, Ci
Electrical Engineering series, El
Mathematics and Computing Machinery series, Ma
Mechanical Engineering series, Me
Physics including Nucleonics series, Ph

For subscription to the complete series or to one or more of the sub-series and for purchase of single copies, please write to

ACTA POLYTECHNICA SCANDINAVICA PUBLISHING OFFICE
Box 5073
Stockholm 5 Phone 61 47 53

This issue is published by
THE DANISH ACADEMY OF TECHNICAL SCIENCES
Copenhagen, Denmark

UDC 534.838.1 : 629.12.07

VERTICAL VIBRATIONS IN CARGO AND PASSENGER SHIPS

Viggo A. Kjær
Professor
Research Laboratory of
Internal Combustion Engines
Technical University of Denmark

ACTA POLYTECHNICA SCANDINAVICA Me 2 (A.P. 244/1958)
Mechanical Engineering Series

SUMMARY

A formula, illustrated by a curve, is given for the permissible displacements dependent on the frequencies of vibration. The ratio of the actual to the permissible displacements is called the vibration factor and vibration characteristics are given showing the vibration factor at important positions of some ships. An advance calculation of the vibration amplitude is developed in case of resonance between the exciting forces from the machinery and one of the natural frequencies of the hull. The calculation is based on an expression for the damping comprising a damping factor found by investigations to vary very little even for ships of rather different dimensions and vibrations of different frequencies. A section deals with vertical vibrations due to torsional vibrations in the crank shaft.

CONTENTS

Page

1. Measurement of vibrations 1

2. Susceptibility of persons to vibrations 4

3. Vibration characteristic. Vibration factor 10

4. Calculation in advance of the maximum vibration amplitude. Damping factor 15

5. Investigation of existing ships 21
   - Ship No 1 23
   - Ship No 2 26
   - Ship No 3 30
   - Ship No 4 32
   - Ship No 5 33

6. Discussion of the examinations in the previous chapter and application of the results 34

7. Vertical vibrations in a ship due to torsional vibrations in the crank shaft 36

8. Final remarks 45

VERTICAL VIBRATIONS IN CARGO AND PASSENGER SHIPS

Viggo A. Kjær
Professor
Research Laboratory of
Internal Combustion Engines
Technical University of Denmark

1. Measurement of vibrations.

The vibrations in ships may be vertical, horizontal or torsional vibrations. Only vertical vibrations are dealt with in this paper.

Vibrations are measured by means of vibrographs. These instruments are built to register vibrations to a convenient scale on a moving strip of paper. The vibrographs are also fitted with a timing device so that the frequency as well as the amplitude of the vibrations may be obtained from the vibrograms.

A vibrograph is a good measuring instrument when it is handled in the right way and the records are interpreted correctly. The amplitudes of vibration are in some degree dependent on the pressure with which the writing point is pressed against the paper. This pressure has to be constant, neither too hard nor too light. In measuring low frequency vibrations the vibrograph should generally be fitted with a supplementary weight. For measuring correctly with the Author's Geiger-vibrograph an additional weight has to be used for frequencies lower than 300 c.p.m. On the other hand, the additional weight is not suitable for measuring vibrations of high frequencies because in this case the registered displacements would be too much reduced. At any rate, the displacements on the graph have to be multiplied both by the nominal magnification and by a correction factor found by experiments. As the result of such experiments the correction factor which has to be used in connection with the Author's Geiger-vibrograph is shown for some cases in Fig. 1 and 2. The correction factor depends on the magnification, the frequency and the amplitude; besides this, on the writing point pressure which, however, may be regarded as constant for the skilled user of a vibrograph.

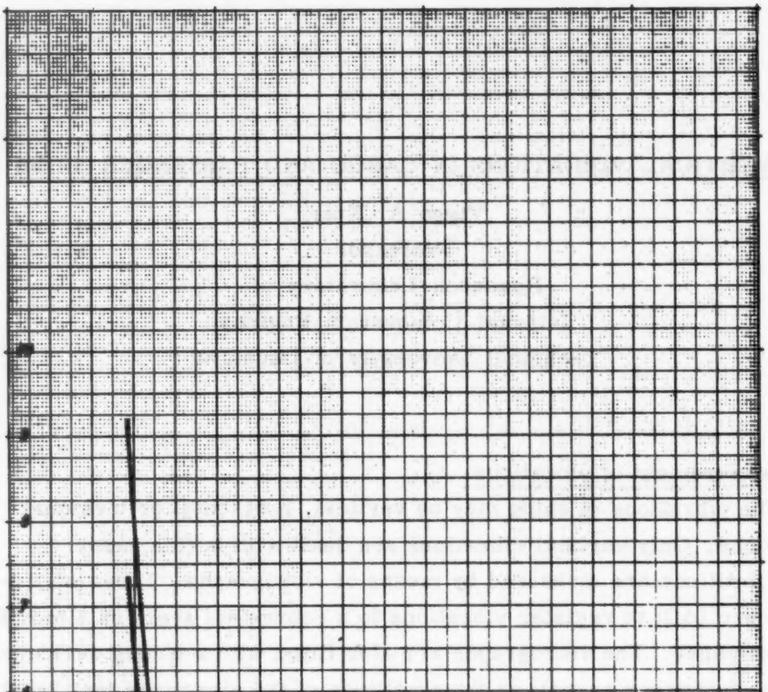

Fig. 1. Correction factor for Geiger vibrograph N⁰ 903.
Magnification 6/1, 2 x amplitude = 3 mm
a light pressure b hard pressure

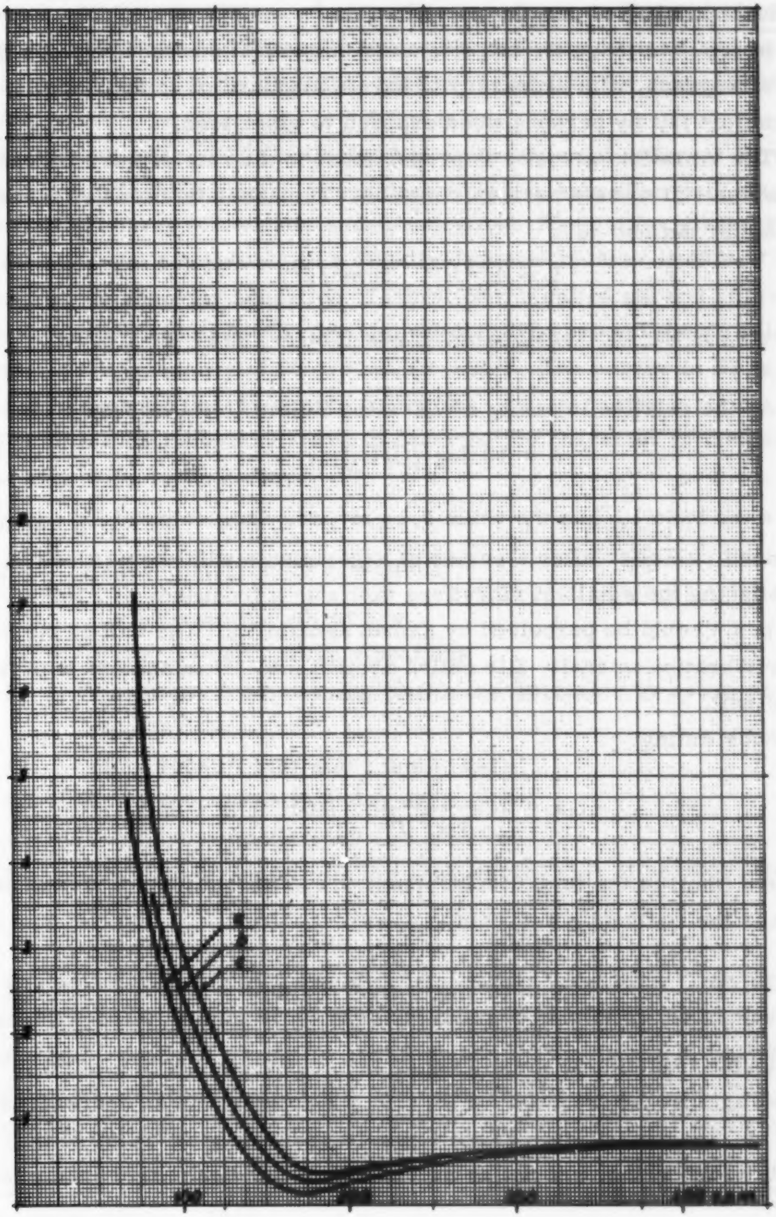

Fig. 2. Correction factor for Geiger vibrograph N° 903.
Magnification 3/1, 2 x amplitude = 0.71 mm
a light pressure b moderate pressure
c hard pressure

2. Susceptibility of persons to vibrations.

In order to judge the importance of the vibrations measured by means of a vibrograph, it is necessary to have a standard of comparison for the discomfort felt by persons when exposed to vibrations.

The vibration intensity is proportional to the rate of work necessary to give unit of mass a harmonic motion. The absolute vibration intensity has Zeller defined as:

$$L = b_o^2/f \qquad (cm^2/s^3)$$

where b_o is the acceleration amplitude (cm/s^2) and f the frequency (c.p.s.)

As the relative intensity of vibration Zeller *) introduced

$$S = 10 \log (L/L_o)$$

where the base L_o has the value $0{,}1\ cm^2/s^3$. This value of L_o was determined by Zeller after a series of experiments with the purpose of establishing the smallest vibrations that with a frequency of 1 Hertz (1 c.p.s.) could be perceived by human sensibility. The unit of the relative vibration intensity S is called decibel (db) **) in analogy with the sound unit.

Substituting

$$a = 20 \frac{b_o}{4\pi^2 f^2} \qquad \text{and} \qquad n = 60\ f$$

where a (mm) is the total displacement, i.e. 2 x amplitude of vibration, and n is the frequency in c.p.m., gives

$$S = 10 \log \frac{a^2 n^3}{5550} \qquad \text{(db)}$$

In a double logarithmic graph (Fig. 3) the double amplitude a (mm) is plotted against the frequency n (c.p.m.) as straight lines corresponding to 0, 10, 20 ..... 80 db. These lines, representing the relative intensity of vibration, are not an expression of the discomfort produced

---

* W. Zeller in an article in Automobiltechnische Zeitschrift 1949 page 95 or Hütte I, 1955, page 610

** The base value L_o was formerly fixed at $0{,}5\ cm^2/s^3$ and the relative vibration intensity in accordance therewith was called pal.

on persons by the vibrations. A classification of human discomfort caused by vibrations of different frequencies is a very difficult and uncertain matter, one of the principal reasons being that the response of different persons to vibrations is not the same.

F. Postlethwaite *) has collected and critically used different series of test results to draw curves of constant vibration sensation in analogy with existing curves of constant sound sensation. The curves given by Postlethwaite are reproduced in a slightly different manner in Hütte **). The unit of the annoying sensation of vibrations is called pal and curves are drawn representing 0, 10, 20 ..... 80 pal. The pal and the db value coincide at the frequency of 60 c.p.m. (= 1 Hz). The curves in Hütte are used in drawing the pal-curves in Fig. 3.

The following table may serve to describe the character of the pal-values according to W. Zeller ***).

Pal	Characterization of vibrations
0 - 10	Just perceptible vibrations
10 - 20	Distinctly perceptible vibrations
20 - 30	Vibrations from passing traffic insupportable by house occupants
30 - 40	Vibrations in easy-going motor cars
40 - 50	Usual vibrations in motor cars
50 - 60	Very annoying vibrations in vehicles
60 - 80	Physical distress. By low frequencies risk of seasickness

The pal-curves in Fig. 3 are of course rather uncertain. They are principally based on laboratory experiments and chiefly on the series of experiments conducted by Reiher and Meister ****). Although these tests are very comprehensive and universally appreciated it is not certain that the results may be transferred directly on the conditions of a ship.

* Engineering 28. jan., 1944.

** Hütte I, 1955, page 612.

*** Automobiltechnische Zeitschrift 1949, page 97.

**** F. S. Meister: Empfindlichkeit des Menschens gegen Erschütterungen, Forschung Bd. 6, Heft 3.

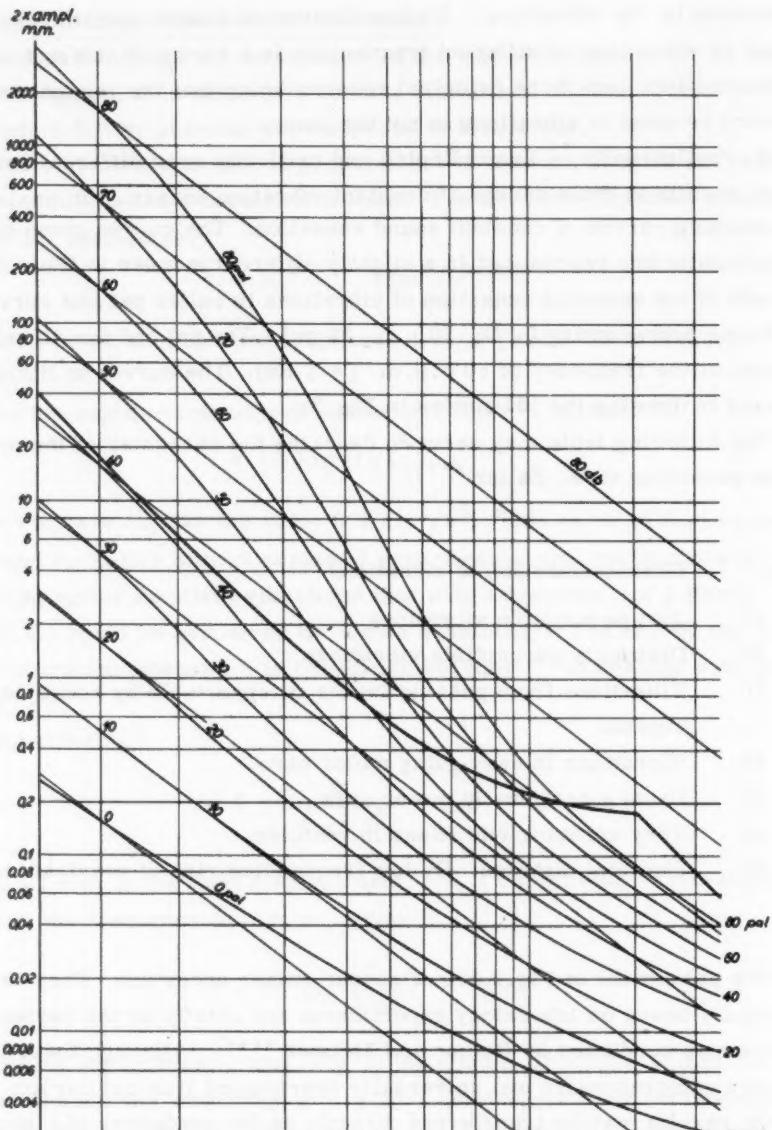

Fig. 3. Decibel- and pal-curves by different frequencies.
The thick drawn curve represents the limit of permissible vibrations according to the Author.

During the experimental determination of the pal-values the attention of the persons subjected to the test is exclusively turned to the vibrations. Aboard, the passengers, even if the vibrations are felt slightly, have their minds occupied by many other things, so that they easily forget to pay attention to the vibrations if these are not of a really severe order.

On experiments carried out on a ship to determine the limit of permissible vibrations not much has been published. S. E. Lundberg *) remarks that vibrations are not annoying when

$$a \cdot n^{1,5} < 1480$$

The best way to determine the limit of permissible vibrations on a ship is to make investigations aboard by means of vibrograms and questions to the crew or passengers about the annoying effect of the vibrations. Based on vibration tests on more than 100 ships and on the comments from the persons aboard and personal reactions, the Author came to the following formulæ for the limit of permissible vibrations of ordinary cargo and passenger ships:

$$(a - 0{,}16)\, n^2 = 64000 \qquad \text{when} \qquad n < 2000 \text{ c.p.m.}$$

$$a\, n^2 = 720000 \qquad \text{when} \qquad n > 2000 \text{ c.p.m.}$$

where a, as above, is the double amplitude in mm. The last formula is deduced from the consideration that the amplitude of the vibration acceleration must not exceed a certain percentage of the gravitational constant g. The formula corresponds to the value of 0.4 g. The two formu-

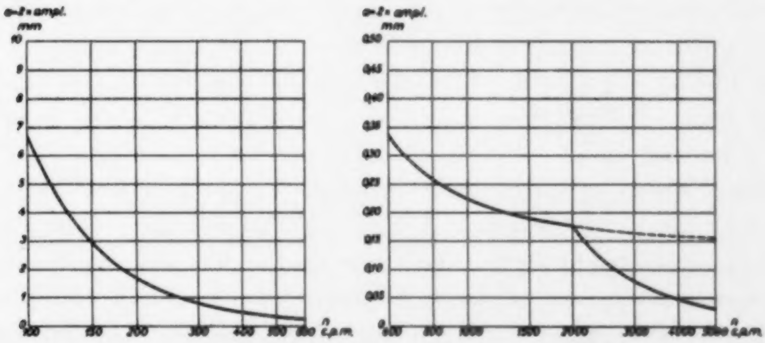

Fig.4. The Author's curve for the limit of permissible vibrations.

Equations of curves

$(a - 0.16)\, n^2 = 64000$ when $n < 2000$

$a\, n^2 = 720000$ when $n > 2000$

* IVA 1938, hefte 4.

lae are shown as the thick curve in Fig.3 and without the use of a logarithmic scale for the ordinates, representing the double amplitude a, in Fig.4 a and 4 b.

It may perhaps be found surprising that the low-frequency end of the limit curve in Fig. 3 corresponds to about 40 pal while the high-frequency end exceeds what Zeller has called 80 pal. The curve is none the less, according to the Author's experiences with ships, the expression of a constant vibration sensation by different frequencies. While many persons can stand low-frequency vibrations of 50 - 60 pal for a short time, for inst. during laboratory tests, the same vibrations may easily cause seasickness if they continue for hours aboard. On the other hand, persons may support, even for a long duration, high-frequency vibrations with greater displacements than corresponding to the curve of 80 pal. According to the Author's experience the feeling of discomfort is practically only dependent on the vibration displacement at frequencies of more than about 1000 c.p.m. *). For 18 years the Author has used the curve in Fig. 4, in which time he has had to do with many ships, but he has not met with a single case that could induce him to change his mind respecting the curve as a reasonable limit for the permissible vibrations of ships.

The curve, however, should not be used for those parts of a ship where no persons are staying, nor for very special ships. On an ice-breaker for inst. it cannot be prevented that much greater vibrations than those indicated in Fig.4 occasionally occur. On the other hand, for a sea-exploring ship, on which the scientifical staff use microscopes and other fine instruments, the limit of permissible vibrations lies sensibly lower than indicated in Fig. 4.

If the vibration is not a harmonic motion but a complex one, consisting of several combined harmonic motions, the constant displacement vibration that gives the same feeling of discomfort should be considered. This is illustrated in Fig. 5, which shows a complex vibration consisting of the superposition of harmonic vibrations with 3, 4 and 6 cycles per revolution of the engine. The varying total displacement has maximum a_{max} and minimum a_o. Every time the maximum amplitude is reached it will be felt like a shock and the complex motion is equivalent to a simple harmonic motion with the double amplitude

---

* Compare the curve of Constant in Engineering 28. jan. 1944

$$a = a_o + 2\,(a_{max} - a_o) = 2\,a_{max} - a_o$$

and with 4 cycles on the length L that is to say per revolution.

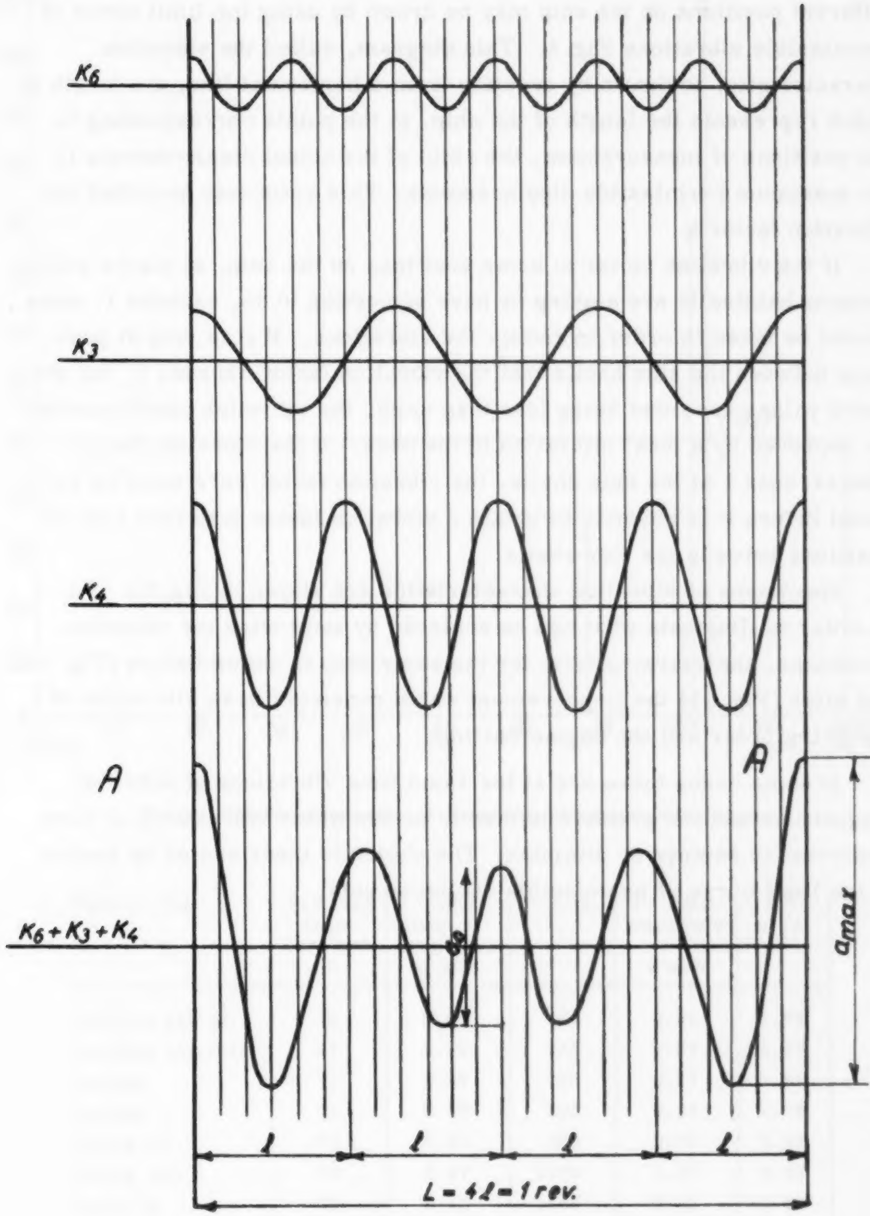

Fig. 5. Complex vibration and its harmonic components.

3. Vibration characteristic. Vibration factor.

After the vibrations have been recorded and the total displacements and the frequencies measured, a diagram illustrating the vibrations at different positions on the ship may be drawn by using the limit curve of permissible vibrations Fig. 4. This diagram, called the vibration characteristic, is drawn by erecting from a horizontal line, the length of which represents the length of the ship, at the points corresponding to the positions of measurement, the ratio of the actual displacements to the maximum permissible displacements. This ratio may be called the vibration factor k.

If the vibration factor at some positions on the ship, at places where persons habitually are staying or have something to do, exceeds 1, steps should be taken in order to reduce the vibrations. If it is only at positions between the side-shells that the vibration factor exceeds 1, the seaboard values recorded being less than unity, the vibration conditions can be improved by a local alteration of the deck. If the vibration factor also exceeds 1 at the side-shells, the vibration factor here must be reduced before it is possible to obtain a vibration factor less than 1 at all positions between the side-shells.

Specimens of vibration characteristics are shown in Fig. 6 a - 6 d. In order to illustrate what can be achieved by improving the vibration conditions, the characteristic for the same ship is shown before (Fig. 6 a) and after (Fig. 6 b) the improvement which consisted in an alteration of the firing order and the engine seating.

In some cases there are at the same time vibrations of different frequencies and the problem is then to decide which vibration it is most important to remove or diminish. The choice is made easier by means of the limit curve of permissible displacements.

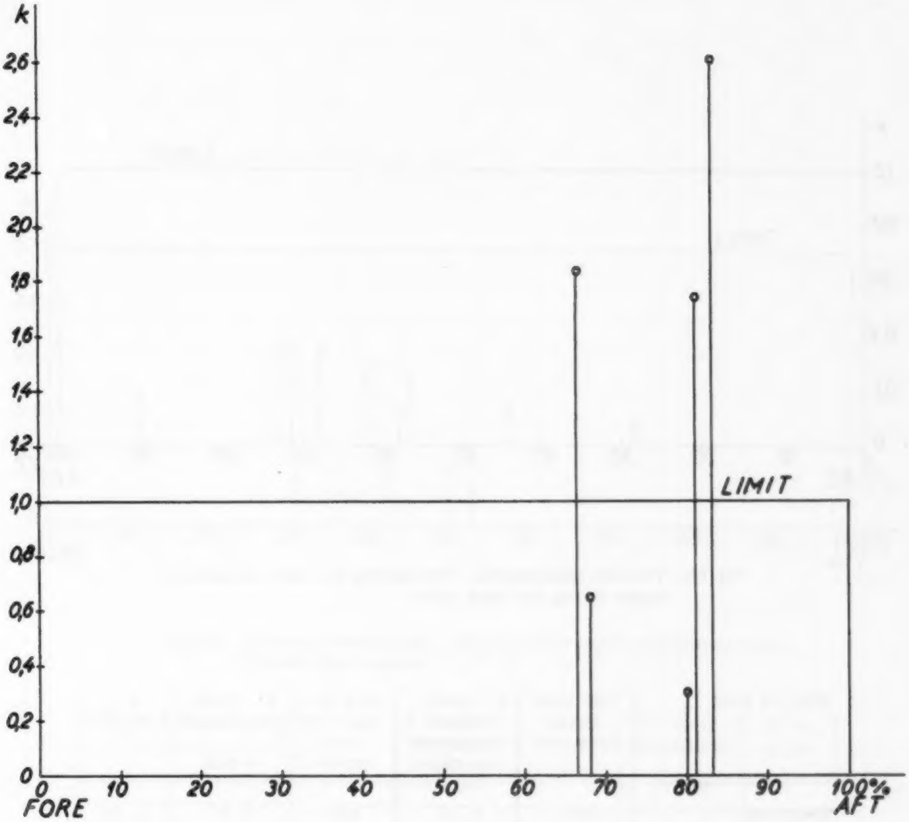

Fig. 6a. Vibration characteristic. Passenger ship with 4-stroke Diesel engine.

Part of ship	Per cent from fore-end	2 · ampl. from vibrograms a_1 mm	c.p.m.	2 · limit amplitude a mm	k = a_1/a
Smoking sal. aft	73	0.90	600	0.34	2.64
Smoking sal. fore	57	0.50	725	0.27	1.85
Lounge	71	0.45	725	0.27	1.66
Lounge	71	0.60	600	0.34	1.76
Dining sal.	70	0.20	290	0.92	0.22
Dining sal.	70	0.07	1160	0.22	0.32
Cabin 40	58	0.15	1160	0.22	0.68

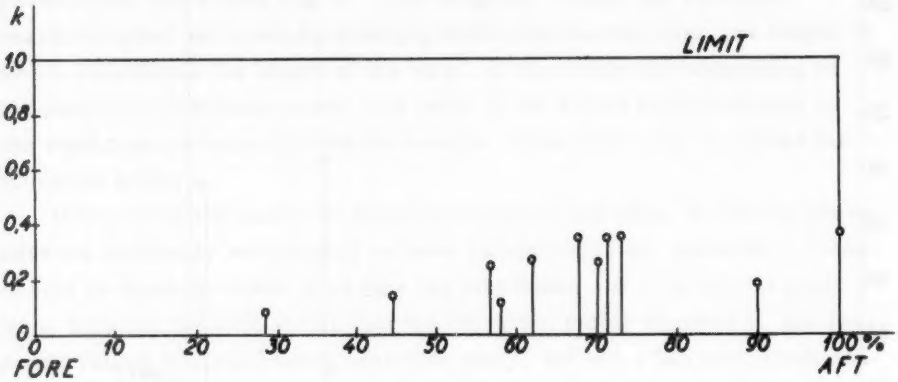

Fig. 6b. Vibration characteristic. The ship Fig. 6 a. after alteration of engine seating and firing order.

Part of ship	Per cent from fore-end	2 · ampl. from vibrograms a_1 mm	c.p.m.	2 · limit amplitude a mm	k = a_1/a
Stern aft	100	0.18	435	0.47	0.38
Stern ship side	90	0.09	435	0.47	0.19
Smoking sal. aft	73	0.08	1000	0.22	0.36
Smoking sal. fore	57	0.05	1300	0.20	0.25
Lounge	71	0.09	725	0.27	0.33
Lounge door	68	0.08	1000	0.22	0.36
Dining saloon	70	0.06	1000	0.22	0.27
Hall	62	0.06	1000	0.22	0.27
Cabin 4	58	0.03	725	0.27	0.11
Cabin 17	45	0.04	725	0.27	0.15
Cabin 26	29	0.02	725	0.27	0.08

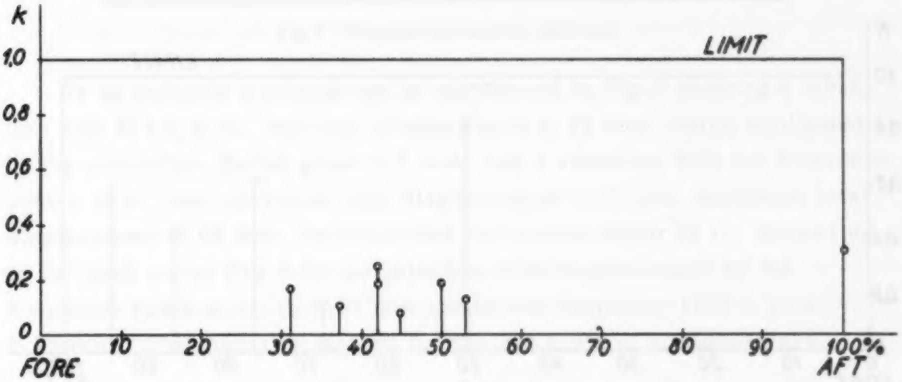

Fig. 6 c. Vibration characteristic. Specially built ship for scientifical work. 4-stroke Diesel engine.

Part of ship	Per cent from fore-end	2 · ampl. from vibrograms a_1 mm	c.p.m.	2 · limit amplitude a mm	k = a_1/a
Aft-end	100	0.075	825	0.25	0.30
Captain	50	0.046	825	0.25	0.18
Laboratory	37	0.017	1650	0.19	0.09
Bridge	31	0.042	825	0.25	0.17
Dining saloon	53	0.030	1650	0.19	0.16
Office	45	0.017	1650	0.19	0.09
Microscope room	42	0.036	1650	0.19	0.19
Table in micr.r.	42	0.045	2475	0.12	0.38

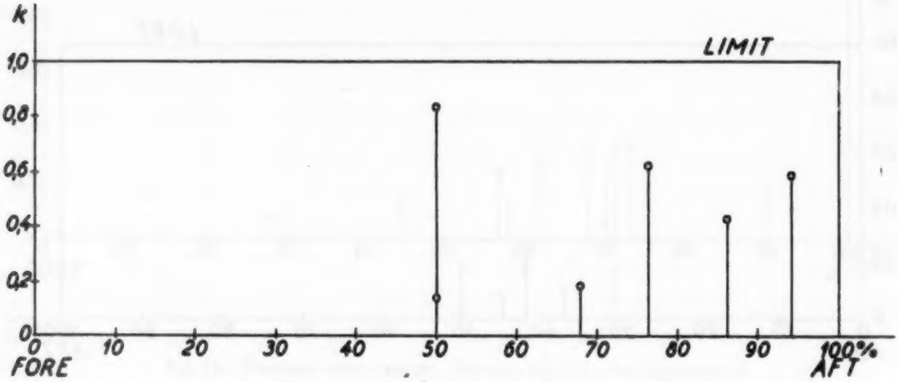

Fig. 6 d. Vibration characteristic. Old steam ferry-boat.

Part of ship	Per cent from fore-end	2 · ampl. from vi-brograms a_1 mm	c.p.m.	2 · limit amplitude a mm	k= a_1/a
Aft-end	95	0.25	464	0.45	0.55
1.class aft-end	86	0.083	1390	0.20	0.41
1.class ladies sal.	77	0.267	464	0.45	0.59
1.class dining sal.	68	0.083	464	0.45	0.18
Captain bridge	50	0.025	2088	0.18	0.14
Bridge (deck)	50	0.33	480	0.44	0.75

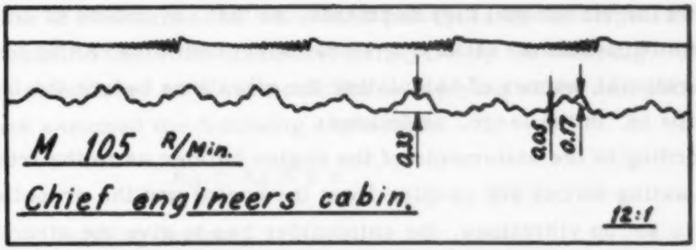

Fig. 7. Vibrogram of complex vibrations.

As an example a vibrogram is reproduced in Fig. 7 showing a vibration with 210 c.p.m. and total displacement 0.33 mm, which multiplied by the correction factor gives 0.5 mm, and a vibration with the frequency 1050 c.p.m. and maximum total displacement 0.17 mm, minimum total displacement 0.05 mm, for which the correction factor is 1. According to the limit curve Fig. 4 the permissible total displacement by the frequency 210 c.p.m. is 1.62 mm and by the frequency 1050 c.p.m. 0.22 mm. The vibration factors for the two kinds of vibrations are

$$k = 0.5 / 1.62 = 0.31 \quad \text{at} \quad 210 \text{ c.p.m.}$$

$$k = (2 \cdot 0.17 - 0.05) / 0.22 = 1,3 \quad \text{at} \quad 1050 \text{ c.p.m.}$$

It is obvious that only little can be obtained by reducing the low-frequency vibration. On the contrary it is very necessary to do something in order to reduce the vibrations with 1050 c.p.m.

It is not possible to give a general rule for the damaging effect of the vibrations upon the structure of the ship. So much is certain, however, that no damages have occurred on the steelships, vibrationally investigated by the Author, in the cases when the vibration factor corresponding to the limit curve in Fig. 4 has been less than 1. The same cannot be said about wooden ships. The fact of keeping below the limit curve is for wooden ships no true insurance against damages.

4. Calculation in advance of the maximum vibration amplitude. Damping factor.

It is not impossible that the day will come when the owners, before taking over a ship, require that the vibrational displacements measured on the trial trip are less than shown by the curve Fig. 4 or any other curve agreed upon to be considered as the limit of permissible vibrations. The problem then arises: has the constructing engineer a method to calculate

beforehand the vibrational displacements, so that, by means of constructive modifications, satisfying vibrational conditions can be obtained.

The rational manner of calculating the vibrations before the building of the ships is, in the large, as follows:

According to the statements of the engine builder as to the frequencies of the pulsating forces and couples from the engine and the propeller which are able to set up vibrations, the shipbuilder has to give the structure supporting the engine, the sternpost and rudder and the decks such dimensions that the natural frequency n_e of these parts differs sufficiently from the frequency n of the exciting forces or couples. The magnification factor given by

$$f = \frac{1}{1 - (n/n_e)^2}$$

must not be too high in the range of normal engine revolutions. When appropriate measures are taken at the beginning, this is neither very difficult nor costly to obtain.

Much more difficult is it to obtain that the 2-, 3- and 4-node natural frequencies of the hull are lying outside the ranges of resonance. Often this problem is impossible to solve. The natural frequencies of the hull cannot be altered and they are in some degree dependent on the conditions of loading. Therefore, the builder of the engine has to do his utmost to reduce to an acceptable value the exciting forces and couples capable of causing vibrations.

The problem is then: is it possible with sufficient accuracy to calculate in advance the maximum vibration amplitudes in case of resonance between the natural hull frequency and the known exciting forces from the machinery. These calculations can be performed, as will now be shown, when knowledge is available about the vibration profile of the hull and the damping at resonance.

Consider first the case when the impulse is a harmonic force applying at an anti-node. The damping is at any rate relatively small in relation to the exciting energy and the vibration motion therefore also harmonic. The internal work done per complete cycle by the harmonic force is

$$\frac{\pi}{2} K_n a$$

where K_n = the amplitude of the force

a = 2 x amplitude of displacement at its point of application.

In case of resonance the work done per cycle by the impulse force must be equal to the work dissipated per cycle by the damping. For the damping, considered as a harmonic force applying at the anti-node, the Author has assumed the following expression

$$K_d = k_d\, A\, a\, n$$

where k_d = damping factor

A = fraction of the area LB, where

L = the length between P.P.

B = the breadth moulded

a = the double amplitude at the anti-node

n = the frequency in c.p.m.

The work dissipated per cycle is then

$$\frac{\pi}{2} k_d\, A\, a^2\, n$$

Putting the work of the impulse force equal to the work dissipated by damping gives

$$a = \frac{Kn}{k_d\, A\, n}$$

It has been confirmed by the Author's calculations of existing ships that k_d only varies slightly. When

K_n is in kg

n in c.p.m.

A in m^2 and $A = 0.3 \cdot L \cdot B$ by 2-node vibrations

$A = 0.36 \cdot L \cdot B$ by vibrations with 3 or more nodes

k_d = 1/45, which is a suitable and sufficiently safe value,

the double amplitude a is found in mm.

The expression above is empirical and in the table page 22 the results from measurements on 5 ships with their corresponding 12 modes of hull vibration are stated. A discussion of the empirical results and their variation will be given later on in this paper.

It is seen that the expression adopted for the damping, supposed to be caused by the movement of the water, shows proportionality between the damping force and the area L·B. It could be expected that the form

and the draught of the ship also influences the damping. From his experience with modern cargo and passenger ships, however, the Author has come to the result that these factors are so little important with regard to the damping that they may be neglected.

In the expressions above the point of application for the harmonic force K_n was an anti-node. If this is not the case with the actual force it will be practical to calculate an equivalent harmonic force K_n, which, applying at the anti-node, produces the same amount of work as the actual force. The expressions above may then be used directly.

If the impulse is a couple M acting at a node of the vibration profile, see Fig. 8, the moment of the couple can be replaced by an equivalent force K_n acting at the anti-node. The vibration profile is supposed to be a sinusoide with the length 2ℓ between the nodes. K_n is then determined by

$$M \frac{\pi}{2} \frac{a}{2} \frac{1}{\ell} = K_n \frac{a}{2}$$

$$K_n = M \frac{\pi}{2\ell}$$

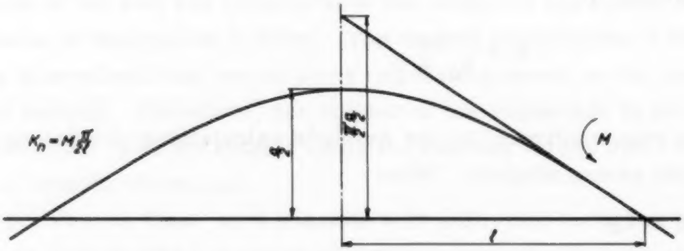

Fig. 8. Harmonic force at anti-node equivalent to harmonic moment at node.

If the moment is not acting at the node or so near the node that the considered part of the vibration profile may be regarded as straight, the amount of work produced by the moment is calculated from the amplitudes at the points where the forces of the couple are acting.

In this connection the crank arrangement of the scavenge pump is of some interest. Generally the crank of the scavenge pump is arranged so that the moment of the primary free force of the pump about the midlength point of the engine has the opposite rotation of the free primary moment of the engine itself. This arrangement corresponds to the Fig. 9.

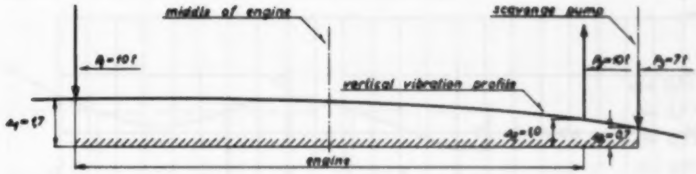

Fig.9. Crank arrangement of scavenge pump.

The free primary moment of the engine is equivalent to a pair of equal and opposite forces P_1 and P_2 each of 10 t. The free primary force of the scavenge pump P_3 = 7 t is opposite to P_2. Acting on the shown vibration profile with the stated amplitudes the work of the impulse forces per complete cycle is

$$(10 \cdot 1.7 - 10 \cdot 1.0 + 7 \cdot 0.7)\pi = 11.9\pi \text{ t cm}$$

If the crank of the scavenge pump is turned 180 degrees the work is reduced to

$$(10 \cdot 1.7 - 10 \cdot 1.0 - 7 \cdot 0.7)\pi = 2.1\pi \text{ t cm}$$

The altered arrangement of the crank improves the vibration conditions when the node is lying outside the engine. On the contrary, if the point of application of P_3 is on the other side of the node, the arrangement of the pump crank corresponding to Fig.9 will be the right one. The position of the nodes can only be calculated approximately beforehand and in some cases it will therefore be very difficult to decide which arrangement of the pump crank is the right one. Moreover, the nodes are not fixed points. Their positions are to some extent dependent on the loading conditions of the ship.

The 2-node natural frequency n_{e1} of the hull can be calculated but the calculation is a long-time job if the result shall be reasonably accurate. A rough guide with respect to n_{e1} only depending on the length of the ship is given in Fig.10

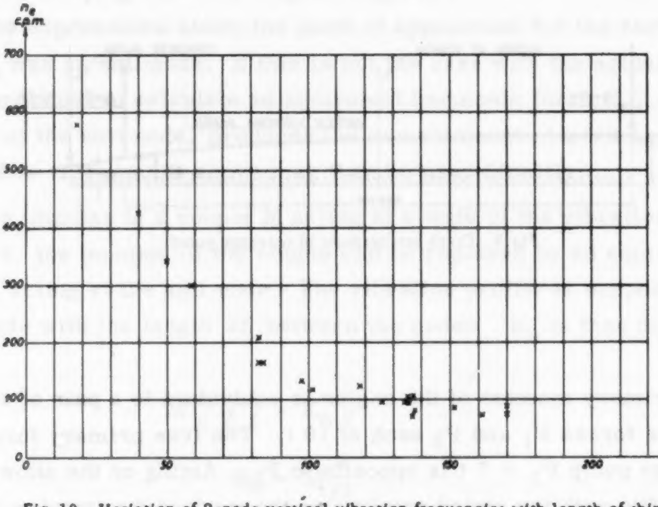

Fig. 10. Variation of 2-node vertical vibration frequencies with length of ship.

The natural frequencies of the hull with more than 2 nodes are even more difficult to calculate exactly. The generally existing relationships between these frequencies are as follows: *)

3 nodes	4 nodes	5 nodes
$n_{e2} = (1.7 - 2.3)\, n_{e1}$	$n_{e3} = (2.6 - 3.2)\, n_{e1}$	$n_{e4} = (3.7 - 5)\, n_{e1}$

Values outside these intervals may occur.
The vibration profiles with 2, 3 or 4 nodes may approximately be drawn by using the statements in Fig. 11.

*** See T. W. Bunyan's article in Transactions of Marine Engineers, april 1955.**

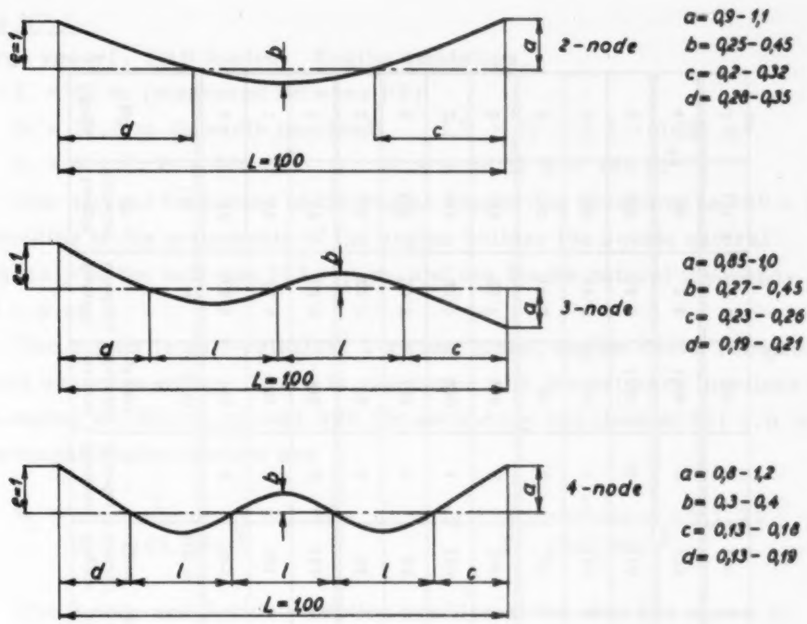

Fig. 11. Vertical vibration profiles.

5. Investigation of existing ships.

This section contains the calculations for some of the ships where the engine-builder's or the Author's investigations have made it possible to determine the damping factor.

It must be borne in mind that the impulses from the engine when transmitted to the hull are multiplied by the magnification factor

$$f = \frac{1}{1 - (n/n_e)^2}$$

where n is the frequency of the impulse force and

n_e is the natural frequency of the structure supporting the engine.

The natural frequency n_e is in the following calculations assumed to be the same whether the impulse from the engine is a moment or a force. In fact, calculations as well as experience show that the natural frequencies of the engine supporting structure in the two cases are practically the same for normally built ships. For ships with a special arrangement of the engine room or of the engine supporting structure this point should, however, be investigated closer.

Ship Nº	Length between PP m	Breadth moulded m	Natural frequency of hull n c.p.m.	Number of nodes	Engine revolutions r.p.m.	Equivalent force at anti-node kg	2 · ampl. at anti-node a mm	Magnification factor f	Damping factor k_d	Permissible 2 · ampl. mm	Vibration factor k	Acceleration ampl. b_0 cm/s^2	Measurements
1	82	12.5	163	2	163	3800	3,2	1.06	1/40	2.60	1.23	47	Engine builder
1	82	12.5	161	2	161	1140	1.05	1.06	1/43	2.60	0.40	15	-
1	82	12.5	282	3	141	1220	0.33	1.20	1/24	0.96	0.34	14	-
2	155	20.7	76	2	76	4000	2.40	1.10	1/40	11.2	0.21	8	-
2	155	20.7	166	3	83	4060	1.0	1.60	1/30	2.48	0.40	15	-
2	155	20.7	220	4	110	8100	2.3	3.0	1/24	1.48	1.55	61	-
2	155	20.7	436	5	109	95000	0.7	1.0	1/37	0.50	1.40	73	-
3	135	18.6	95	2	95	8100	4.6	1.05	1/39	7.25	0.64	23	Author
3	135	18.6	230	3	115	6700	1.8	1.43	1/39	1.37	1.31	52	-
3	135	18.6	1035	11	115	16000	0.12	0.20	1/35	0.22	0.55	70	-
4	18	2.8	1150	2	575	682	1.34	1.05	1/32	0.21	6.38	968	-
5	101.5	15.5	220	3	110	2470	1.0	1.20	1/42	1.48	0.68	27	-

Ship N° 1

Cargo vessel. Half loaded. Engine amidships.

$L = 82$ m (measured between PP)

$B = 12.5$ m (breadth moulded). $LB = 82 \cdot 12.5 = 1025\ m^2$

$A = 0.3\ L\ B = 308\ m^2$ $A = 0.36\ L\ B = 369\ m^2$

The natural frequency of the engine supporting structure is 680 c.p.m. According to the statements of the engine builder the 2-node natural frequency of the hull was 163 c.p.m. and the 3-node natural frequency 282 c.p.m.

The engine is an 8-cylinder, 2-stroke Diesel engine with a reciprocating scavenge pump. There is resonance with the primary impulses of the engine at 163 r.p.m. and with the secondary impulses at 141 r.p.m. The magnification factors are

$$f_1 = \frac{1}{1 - (163/680)^2} = 1.06 \qquad f_2 = \frac{1}{1 - (282/680)^2} = 1.20$$

The 2-node and 3-node vibration profiles of the ship are shown in Fig.12 together with the engine. The 2-node vibration profile with the double amplitude 7 mm at the position "A" aft is the reproduction of the profile drawn by the engine builder on the base of his vibrograms. The 3-node vibration profile has been drawn on the base of two stated displacements measured by the engine builder, one with the double amplitude 0.5 mm at the position "A" and the other with the double amplitude 0.85 mm at the extreme stern.

With reference to the mid-length point of the engine the following impulses were originally acting

a free primary force	=	3,8	t
a free primary moment	=	4.8	tm
a free secondary force	=	1.0	t
a free secondary moment	=	14.25	tm

In the case of resonance at 163 r.p.m. the exciting force $K_n = 1.06 \cdot 3800$ kg is acting practically at the anti-node of the 2-node vibration profile. The double amplitude at this point is 3.2 mm. The free primary moment produces no work. The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{3800 \cdot 1,06}{308 \cdot 3.2 \cdot 163} = 1/40$$

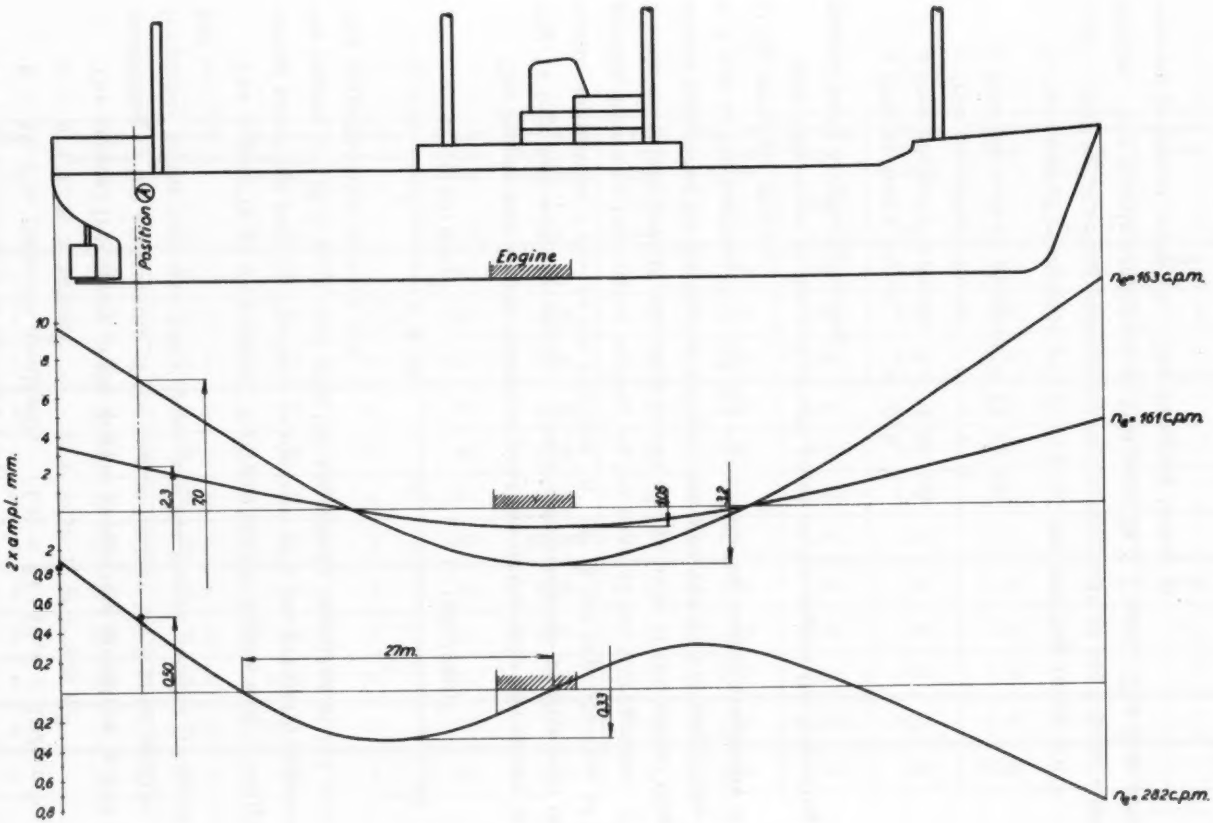

Fig. 12. Vertical vibration profiles ship N° 1.

The permissible double amplitude at 163 c.p.m. is, according to the curve Fig.4, 2.6 mm. The vibration factor corresponding to the harmonic vibration with the double amplitude 3.2 mm is

$$k = 3.2/2.6 = 1.23$$

This is the midships value. At the stern the k-value is greater. These vibrations were a just cause for complaint. By means of balance weights arranged on the crossheads of two cylinders the free primary force was reduced to 1140 kg. The double amplitude at the position "A" aft was reduced to 2.3 mm (Fig.13). The double amplitude at the anti-node becomes $3.2 \cdot 2.3/7 = 1.05$ mm. The natural frequency of the hull, which varies with the loading conditions, was 161 c.p.m. when the new vibrograms were taken.

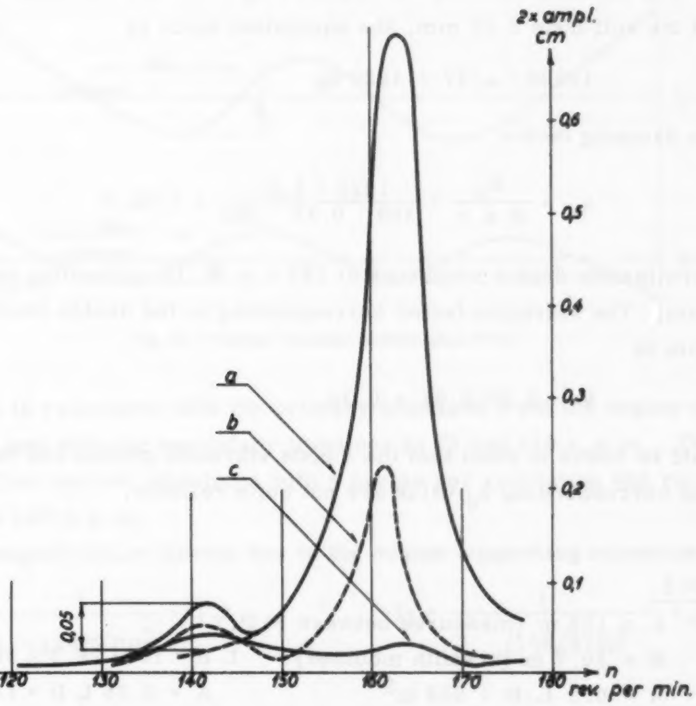

Fig.13. Measured vibration amplitudes ship N° 1, position A in Fig.12.

(a) Free force at anti-node 3800 kg.

(b) Balance weights on crossheads. Free force at anti-node reduced to 1140 kg.

(c) Heavier balance weights on crossheads. Free force reduced to zero.

The damping factor by resonance is now

$$k_d = \frac{K_n}{A\ a\ n} = \frac{1140 \cdot 1.06}{308 \cdot 1.05 \cdot 161} = 1/43$$

The vibration factor corresponding hereto is $k = 1.05/2.6 = 0.4$. At the stern the vibrations were still disturbing. Greater balance weights were therefore arranged and the free primary force was reduced to zero.

In the case of resonance at 141 r.p.m. the free secondary moment is acting on the approximately straight part of the 3-node vibration profile at a node. The free secondary moment, which is practically the same after the application of the balance weights, has at 141 r.p.m. the value

$$14.25 \cdot (141/163)^2 = 10.5 \text{ tm}$$

This moment may be replaced by an equivalent force acting at the anti-node. The distance between the nodes being 27 m and the double amplitude at the anti-node 0.33 mm, the equivalent force is

$$10500 \cdot \pi/27 = 1220 \text{ kg}$$

and the damping factor

$$k_d = \frac{K_n}{A\ a\ n} = \frac{1220 \cdot 1.2}{369 \cdot 0.33 \cdot 282} = 1/23.5$$

The permissible double amplitude at 282 c.p.m. is, according to Fig.4, 0.96 mm. The vibration factor corresponding to the double amplitude 0.33 mm is

$$k = 0.33/0.96 = 0.34$$

It should be borne in mind that the 3-node vibration profile and therefore also the corresponding k_d-value are not quite reliable.

Ship N° 2.

Tanker. $L = 155$ m (measured between P.P.)

$B = 20.7$ m (breadth moulded) $\quad L\ B = 155 \cdot 20.7 = 3210\ m^2$

$A = 0.3\ L\ B = 963\ m^2 \quad A = 0.36\ L\ B = 1155\ m^2$

The natural frequency of the engine supporting structure is 270 c.p.m. The natural frequencies corresponding to the 2-node, 3-node, 4-node and 5-node vibration profile was found by the engine builder to respectively 76, 166, 220 and 436 c.p.m. The frequency 220 c.p.m. is for loaded ship,

the other frequencies are for ship in ballast condition. The vibration profiles shown in Fig. 14 are reproductions of the vibration profiles measured by the engine builder. The engine is an 8-cylinder, 2-stroke, Diesel engine with a reciprocating scavenge pump. The aft-end installation of the engine is shown in Fig. 14

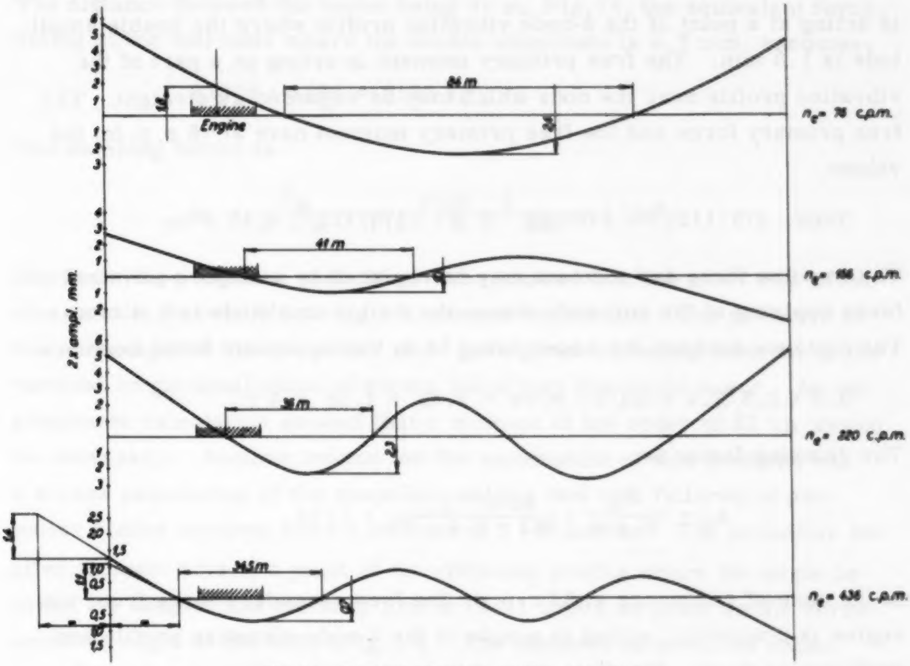

Fig. 14. Vertical vibration profiles, ship N° 2.

Fig. 14.

There is resonance with the primary impulses from the engine at 76 r.p.m. and with the secondary impulses at 83 and 110 r.p.m. There are propeller excited vibrations with 4 cycles per revolution and resonance at 109 r.p.m.

The magnification factors due to the engine supporting structure are

$$f_1 = \frac{1}{1 - (76/270)^2} = 1.1 \qquad f_2 = \frac{1}{1 - (166/270)^2} = 1.6$$

$$f_3 = \frac{1}{1 - (220/270)^2} = 3$$

With reference to the mid-length point of the engine the impulses are as follows:

a free primary force	=	7.6	t
a free primary moment	=	97	tm
a free secondary force	=	1.04	t
a free secondary moment	=	102	tm

In the case of resonance at 76 r.p.m. the exciting free primary force is acting at a point of the 2-node vibration profile where the double amplitude is 1.8 mm. The free primary moment is acting on a part of the vibration profile near the node which may be regarded as straight. The free primary force and the free primary moment have at 76 r.p.m. the values

$$7600 \cdot (76/115)^2 = 3300 \text{ kg} \qquad 97 \cdot (76/115)^2 = 42.3 \text{ tm}$$

The impulse force and moment may be replaced by a single equivalent force applying at the anti-node where the double amplitude is 2.4 mm. The distance between the nodes being 84 m the equivalent force becomes

$$3.3 \cdot 1.8/2.4 + 42.3 \cdot \pi/84 = 2.42 + 1.58 = 4 \text{ t}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{4000 \cdot 1.1}{963 \cdot 2.4 \cdot 76} = 1/40$$

In the case of resonance at 83 r.p.m. the free secondary force from the engine is practically acting at a node of the 3-node vibration profile and produces no work. The free secondary moment is acting on a part of the vibration profile that practically may be regarded as straight. At 83 r.p.m. the free secondary moment is reduced to

$$102 \cdot (83/115)^2 = 53 \text{ tm}$$

The distance between the nodes being 41 m, the equivalent force applying at the anti-node, where the double amplitude is 1.0 mm, becomes

$$53000 \cdot \pi/41 = 4060 \text{ kg}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{4060 \cdot 1.6}{1155 \cdot 1.0 \cdot 166} = 1/30$$

In the case of resonance at 110 r.p.m. the free secondary force from the engine is acting at a node of the 4-node vibration profile and produces no

work. The free secondary moment is acting on a part of the vibration profile that practically may be regarded as straight. At 110 r.p.m. the free secondary moment is reduced to

$$102 \cdot (110/115)^2 = 93 \text{ tm}$$

The distance between the nodes being 36 m, Fig. 14, the equivalent force, acting at the anti-node where the double amplitude is 2.3 mm, becomes

$$93000 \cdot \pi/36 = 8100 \text{ kg}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{8100 \cdot 3}{1155 \cdot 2.3 \cdot 220} = 1/24$$

Marks on the propeller shaft from wear showed that the shaft was lifted clear of the sterntube bearing 4 times per revolution. The pressure of the water against the propeller must have given a bending moment in the vertical longitudinal plane of such a value that this could occur. An approximate calculation showed that a moment of the order of 82 tm should be necessary. Another reason for the assumption of this moment was a stress calculation of the propeller and the fact that failures of propeller blades occured after a service of 3 - 6 months. The propeller excited moment acts at a point of the vibration profile where the angle between the tangent and the base line is 14/11 times as great as the corresponding angle at the node (Fig.14). The distance between the nodes being 34.5 m, the equivalent force, replacing the propeller moment and acting at the anti-node where the double amplitude is 0.7 mm, becomes

$$K_n = 82000 \frac{14}{11} \frac{\pi}{34.5} = 95000 \text{ kg}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{95000}{1155 \cdot 0.7 \cdot 436} = 1/37$$

The permissible double amplitude at 76, 166, 220 and 436 c.p.m. are, according to the limit curve Fig.4, respectively 11.2, 2.48, 1.48 and 0.50 mm. The vibration factors corresponding to the double amplitudes 2.4, 1.0, 2.3 and 0.7 mm are respectively

$k = 2.4/11.2 = 0.21$ $\quad\quad$ $k = 1.0/2.48 = 0.40$

$k = 2.3/1.48 = 1.55$ $\quad\quad$ $k = 0.7/0.5 = 1.40$

Ship Nº 3

Cargo vessel. Engine amidships.

$L = 135$ m (measured between P.P.)

$B = 18.6$ m (breadth moulded) $\quad L\ B = 135 \cdot 18.6 = 2510\ m^2$

$A = 0.3\ L\ B = 753\ m^2 \quad A = 0.36\ L\ B = 904\ m^2$

The natural frequency of the engine supporting structure is 420 c.p.m. The 2-node natural frequency of the hull is 95 c.p.m. and the 3-node natural frequency 230 c.p.m. These frequencies are found by the measurements of the Author. The engine is a 7-cylinder, 2-stroke, Diesel engine with a reciprocating scavenge pump. There is resonance with the free primary impulses from the engine at 95 r.p.m. and with the free secondary impulses at 115 r.p.m. The magnification factors due to the engine supporting structure are

$$f_1 = \frac{1}{1 - (95/420)^2} = 1.05 \qquad f_2 = \frac{1}{1 - (230/420)^2} = 1.43$$

At 115 r.p.m. the free impulses from the engine have the following values:

a free primary force	=	11.9 t
a free primary moment	=	23.6 tm
a free secondary force	=	1.0 t
a free secondary moment	=	109 tm

In the case of resonance at 95 r.p.m. the free primary force is reduced to

$$11900 \cdot (95/115)^2 = 8100 \text{ kg}$$

The point of application of this force is approximately the anti-node with the double amplitude 4.6 mm. The free primary moment produces no work. The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{8100 \cdot 1.05}{753 \cdot 4.6 \cdot 95} = 1/39$$

Resonance at 115 r.p.m. is now considered. A part of the 3-node vibration profile is shown in Fig.15 where also the installation of the engine may be seen. The free secondary moment is replaced by 2 equal and opposite forces applying at the fore-end and aft-end of the engine. The distance between them being 10 m, the forces are each of 109/10 = 10.9 t.

The double amplitudes at the points of application are 0 and 1.1 mm. The work produced by the free secondary force is so small that it may be neglected. The equivalent force acting at the antinode, where the double amplitude is 1.8 mm, becomes

$$10.9 \cdot 1.1/1.8 = 6.7 \text{ t}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{6700 \cdot 1.43}{904 \cdot 1.8 \cdot 230} = 1/39$$

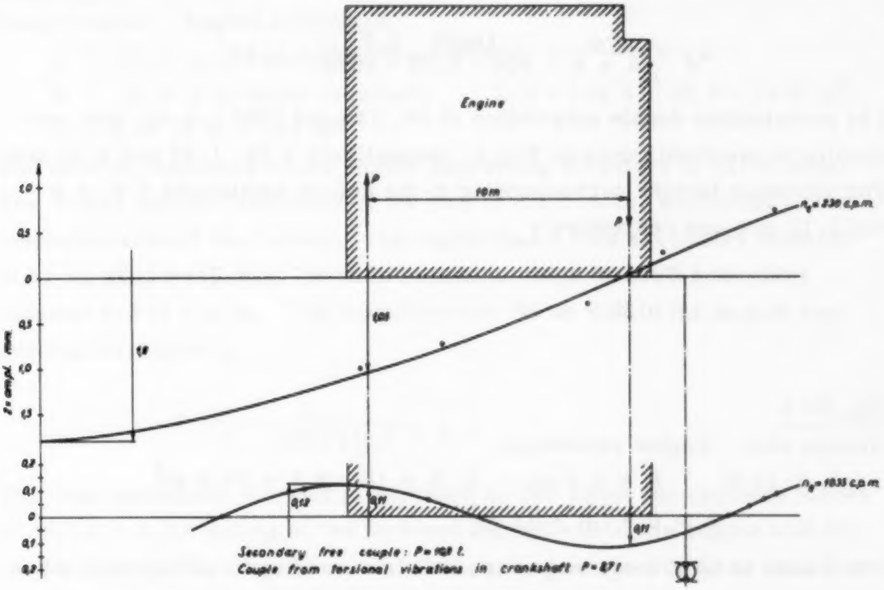

Fig.15. Vertical vibration profiles, ship N° 3.

Torsional vibrations in the crankshaft at 115 r.p.m. occasion, as it will be shown later on, besides a free force a nonbalanced free moment of 87000 kgm pulsating with 9 · 115 = 1035 c.p.m. and another moment of the same value with 11 · 115 = 1265 c.p.m. The vibrograms taken by the Author indicate that the 11-node natural frequency of the hull was 1035 c.p.m. The corresponding vibration profile is shown in Fig.15. The impulse force is relatively small and applies practically at a node so that its energy-producing effect may be neglected. The impulse moment is replaced by two equal and opposite forces of 8700 kg. The double amplitudes at the points of application are both 0.11 mm. The

magnification factor due to the engine supporting structure is

$$f = \frac{1}{1 - (1035/420)^2} = 0.2$$

The equivalent force acting at the anti-node, where the double amplitude is 0.12 mm, becomes

$$8700\ (0.11 + 0.11)/0.12 = 16000\ \text{kg}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{16000 \cdot 0.2}{904 \cdot 0.12 \cdot 1035} = 1/35$$

The permissible double amplitudes at 95, 230 and 1035 c.p.m. are, according to the limit curve in Fig.4, respectively 7.25, 1.37 and 0.22 mm. The vibration factors corresponding to the double amplitudes 4.6, 1.8 and 0.12 mm are respectively

$$k = 4.6/7.25 = 0.64 \qquad k = 1.8/1.37 = 1.31$$

$$k = 0.12/0.22 = 0.55$$

Ship No 4

Wooden ship. Engine amidships.

$$L = 18\ \text{m.} \qquad B = 2.8\ \text{m.} \qquad L\ B = 18 \cdot 2.8 = 50.4\ \text{m}^2$$

$$A = 0{,}3\ LB = 15\ \text{m}^2$$

The 2-node natural frequency of the hull is according to vibrograms taken by the Author 1150 c.p.m. There is resonance with the free secondary impulses from the engine at 575 r.p.m. ($\omega = 60\ s^{-1}$). The engine is a 4-cylinder, 4-stroke, Diesel engine, bore 180 mm, stroke 270 mm ($r = 13.5$ cm) with all the cranks in one plane. The reciprocating weights are $G = 68.8$ kg. The crank/connection rod ratio $\lambda = 1/5$. The free secondary force is calculated at

$$\frac{G}{g}\, r\,\omega^2\,\lambda = \frac{68.8}{9.81}\, 13.5 \cdot 3600 \cdot \frac{1}{5} = 682\ \text{kg}$$

The magnification factor is $f = 1.05$. The point of application of the impulse force is the anti-node with the double amplitude 1.34 mm. The damping factor becomes

$$k_d = \frac{K_n}{A\ a\ n} = \frac{682 \cdot 1.05}{15 \cdot 1.34 \cdot 1150} = 1/32$$

The permissible double amplitude at 1150 c.p.m. is, according to the limit curve in Fig.4, 0.21 mm. The vibration factor corresponding to the double amplitude 1.34 mm is

$$k = 1.34/0.21 = 6.38.$$

Ship N° 5

Cargo vessel. Engine amidships.

L = 101.5 m (measured between P.P.)

B = 15.5 m (breadth moulded) $L\ B = 101.5 \cdot 15.5 = 1575\ m^2$

$A = 0.36\ L\ B = 567\ m^2$

The natural frequency of the engine supporting structure is 535 c.p.m. The 3-node natural frequency of the hull is 220 c.p.m. according to the measurements of the Author. The engine has a free secondary moment of 42 tm at 110 r.p.m. There is resonance with the free secondary moment at 110 r.p.m. The magnification factor due to the engine supporting structure is

$$f = \frac{1}{1 - (220/535)^2} = 1.2$$

The free secondary moment is replaced by two equal and opposite forces of 42/11 = 3.8 t acting at the fore-end and aft-end of the engine with the length 11 m. The vibration profile which is shown in Fig.16 is based on the measurements of the Author. The double amplitudes at the points of application of the forces are 0.95 and 0.3 mm. The equivalent force acting at the anti-node where the double amplitude is 1 mm becomes

$$3800\ (0.95 - 0.3)/1 = 2470\ \text{kg}$$

The damping factor is

$$k_d = \frac{K_n}{A\ a\ n} = \frac{2470 \cdot 1.2}{567 \cdot 1 \cdot 220} = 1/42$$

The permissible double amplitude at 220 c.p.m. is, according to the limit curve in Fig.4, 1.48 mm. The vibration factor corresponding to the double amplitude 1 mm is $k = 1/1.48 = 0.68$.

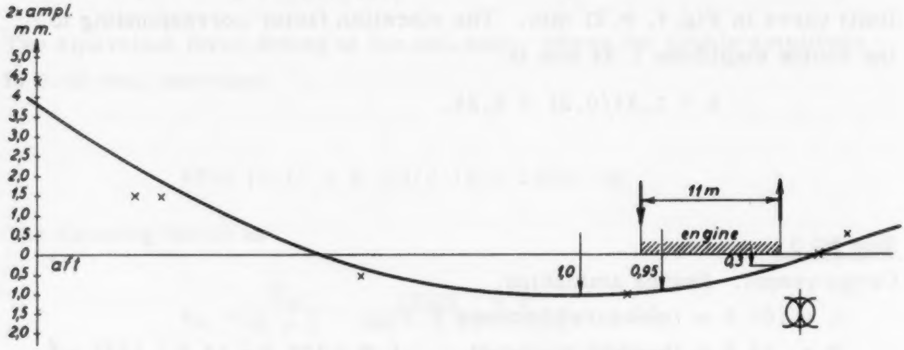

Fig. 16. Vertical vibration profiles, ship N° 5.

6. Discussion of the examinations in the previous chapter and application of the results.

The examinations of the five ships are, as previously mentioned, resumed in the table page 22. It will be noted that the calculated values of the damping factor k_d are not varying very much. Indeed, the values are lying in the range between 1/43 and 1/24.

The obtained k_d-values, of course, contain some uncertainty. This is partly due to the difficulties on the trial trip connected with the taking of a sufficient number of vibrograms just at resonance permitting to determine the maximum vibration amplitudes. Partly it is due to the fact that a ship, even if the loading conditions are identical and the engine is working in practically the same manner, may have vibrations of a more or less severe order.

These variations must be due to the water conditions. Very special cases as the sailing in canals or in shallow water will not be considered here. It is supposed that the ship is moving in the open sea. But even in this case it has often been noted that the vibrations are greatest when the ship is sailing in absolutely calm water for instance in the deep and generally very calm Norwegian fiords. On the contrary, the vibrations may entirely disappear when the ship is sailing in rough water.

This may be explained directly by a varying damping effect of the water but it may also be due to an irregularity in the number of engine revolutions which is greater when the sea is rough than when it is calm. The impulses require a certain time to set up the maximum vibration amplitudes. A rough calculation for the ship Nº 3 showed that it would take about 20 seconds for the free secondary moment to set up the maximum vibrations when the number of engine revolutions per minute was assumed to be constant. In fact, the engine speed is, even in calm water, never quite constant which may be explained either by slight swells or by small movements of the rudder. In a case where the vibrations are getting too severe in calm water it should be possible to prevent this by giving the engine an intentional speed variation. An arrangement for obtaining this may be of a very simple design, whether a mechanical, an electric or a hydraulic device is used.

As previously mentioned, it is almost impossible to prevent resonance with all the natural frequencies of the hull. This is partly due to the uncertainty of the calculations - especially the calculations of the higher frequencies which become complicated by the influence of local vibrations - partly to the variation of the natural frequencies caused by the loading conditions. Often this variation amounts to 10 or 15 percent.

Therefore, it is often necessary in the advance calculation of the vibration amplitudes to assume that resonance is occurring, and if the form of the vibration profiles are exactly calculated or approximately known from ships similar to the ship being designed, the ship builder has, in the calculations given above, the means to examine if the free forces or couples of the engine, stated by the engine builder, produces vibrations which may be regarded as permissible or that the vibrations will be of such an order that a better balancing of the engine is needed. In order to keep well within the limit it will of course be necessary to apply the smallest of the values found for k_d. According to the investigations of existing ships it seems reasonable to put $k_d = 1/45$ as a sufficiently safe value without coming too much on the safe side.

If the natural frequencies of the hull and the form of the vibration profiles are not accurately known, it will at any rate be possible, by means of the approximate values stated on page 19, to make such calculations that the character of the vibrations may be estimated. If unpermissible vibrations are to be expected, as accurate a calculation as possible of the natural frequencies and the position of the nodes should

be made before a better balancing of the engine is demanded.

It is often possible to build 4-stroke Diesel engines without free forces and couples in the vertical longitudinal plane. This is not the case with regard to 2-stroke engines. For many reasons it is desired that the torque is as uniform as possible and to obtain this it will be necessary to put up with an imperfect balancing of the engine. There will be free forces or couples with a frequency equal to or the double of the number of engine revolutions n_m. Impulses of the frequency n_m or $2\ n_m$ may also be due to a damaged or a badly balanced propeller. Furthermore, the propeller causes impulses with a frequency equal to the number of blades multiplied by n_m. These impulses may be important when the aperture clearances between the propeller and the stern or the rudder are too small *). An example of this was ship Nº 3.

It is already well known that the above mentioned impulses may set up vibrations. It is less known that torsional vibrations in the crank shaft may produce vertical vibrations in the hull **). This was the case with ship Nº 3. In the next section it will be shown how the impulse couple used in the calculation of ship Nº 3 has been found.

7. Vertical vibrations in a ship due to torsional vibrations in the crank shaft.

Fig. 17 shows the form of torsional vibrations in the shafting with two nodes corresponding to the amplitude 1 at the scavenge pump. The natural frequency is 1139 c.p.m. The angular frequency is $\omega = \pi \cdot 1139/30 = 119\ s^{-1}$. The actual amplitudes measured on the crank radius are stated in the table page 41. These amplitudes correspond to an additional stress of $\pm$ 240 kg/cm^2 in the middle of the crank shaft.

---

* Institute of Marine Engineers, Transactions, April 1955

** C. Hösch: Die Überwichte serienmässiger Verbrennungsmotoren. Maschinenbautechnik, Juni 1955.

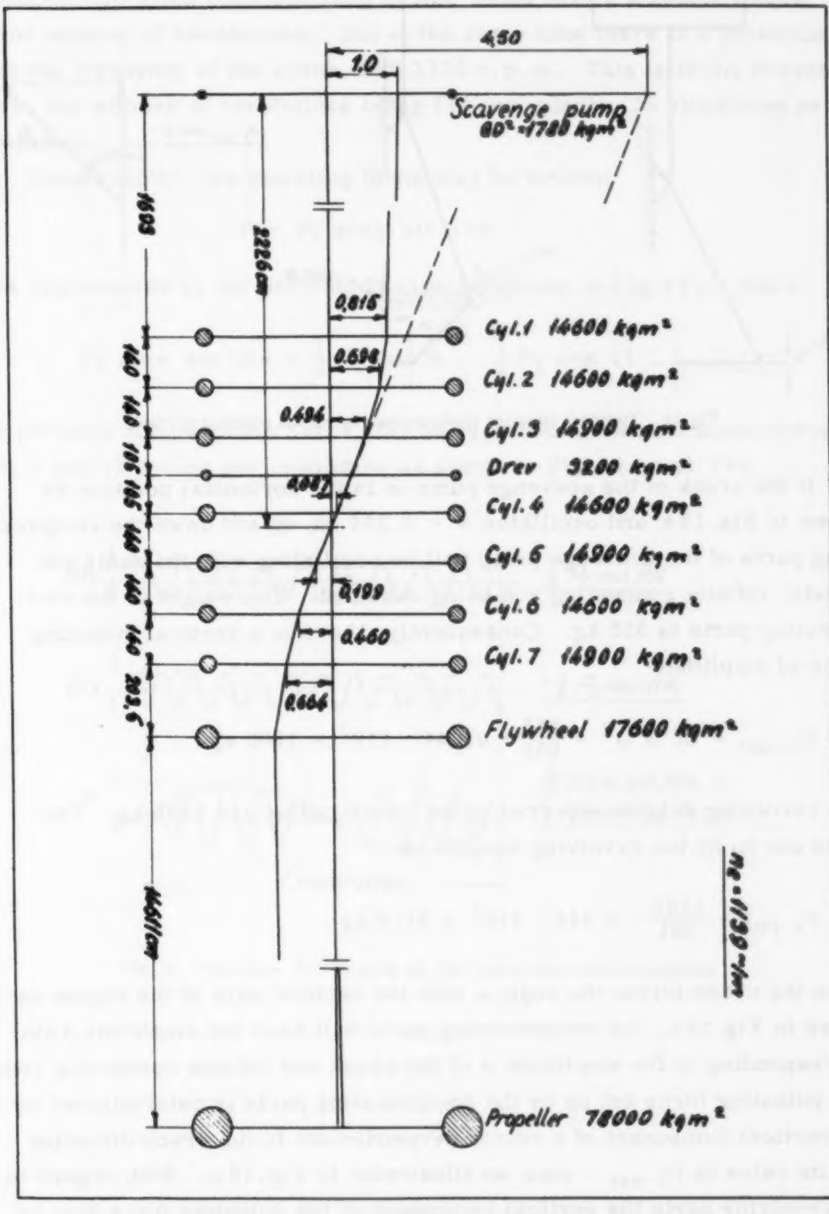

Fig. 17. Two-node form of torsional vibration in the shafting ship N° 3. n_e = 1139 c.p.m.

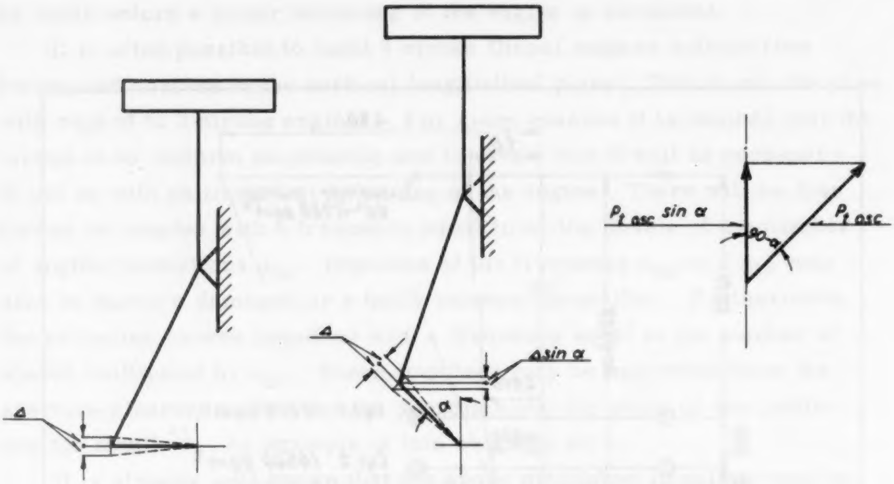

Fig. 18. Torsional vibration displacements by various positions of crank.

If the crank of the scavenge pump is in the horizontal position as shown in Fig. 18 a and oscillates Δ = 0.244 cm up and down the reciprocating parts of the scavenge pump will be oscillating with the same amplitude, infinite connecting rod being assumed. The weight of the reciprocating parts is 555 kg. Consequently, there is a vertical pulsating force of amplitude

$$P_{t\ osc} = m\ \Delta\ \omega^2 = \frac{555}{981} \cdot 0.244 \cdot 119^2 = 1960\ \text{kg}$$

The revolving weights referred to the crank radius are 1165 kg. The force set up by the revolving weights is

$$P_{t\ rot} = \frac{1165}{981} \cdot 0.244 \cdot 119^2 = 4110\ \text{kg}$$

When the crank forms the angle α with the vertical axis of the engine as shown in Fig. 18 b, the reciprocating parts will have the amplitude Δ sin corresponding to the amplitude Δ of the crank and infinite connecting rod. The pulsating force set up by the reciprocating parts is determinated by the vertical component of a vector perpendicular to the crank direction and its value is $P_{t\ osc} \cdot \sin\alpha$ as illustrated in Fig. 18 c. With regard to the revolving parts the vertical component of the pulsating force may be represented in a similar way. The vertical component is $P_{t\ rot} \cdot \sin\alpha$ while the horizontal component is $P_{t\ rot} \cdot \cos\alpha$..

It is seen that the pulsating vertical force, due to the torsional vibrations, twice per revolution reaches its greatest numerical value. A single harmonic vibration corresponding hereto would have a frequency equal to the number of revolutions. But at the same time there is a pulsation with the frequency of the crank shaft 1139 c.p.m. This is in the present case, the number of revolutions being 114 per minute, 10 vibrations per revolution.

Consequently, the pulsating force may be written

$$P = P_t \sin\alpha \sin 10\alpha$$

It is represented by the periodic fluctuation shown in Fig. 19 c. Since

$$P_t \sin\alpha \sin 10\alpha = \frac{1}{2} P_t \cos 9 \quad - \frac{1}{2} P_t \cos 11$$

the periodic non-harmonic force may be replaced by two harmonic forces with 9 and 11 cycles per revolution as shown in Fig. 19 a and 19 b.

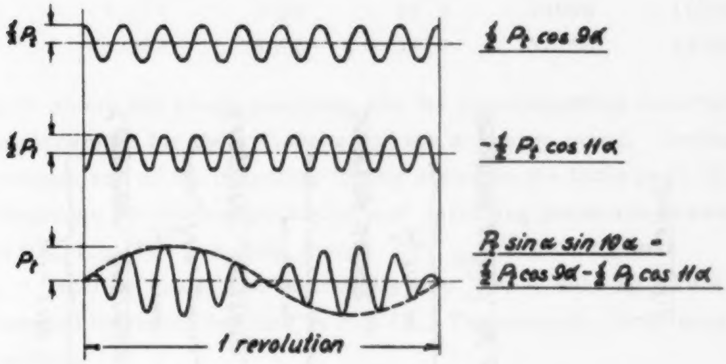

Fig. 19. The curve $P = P_t \sin\alpha \sin 10\alpha$ and its harmonic components.

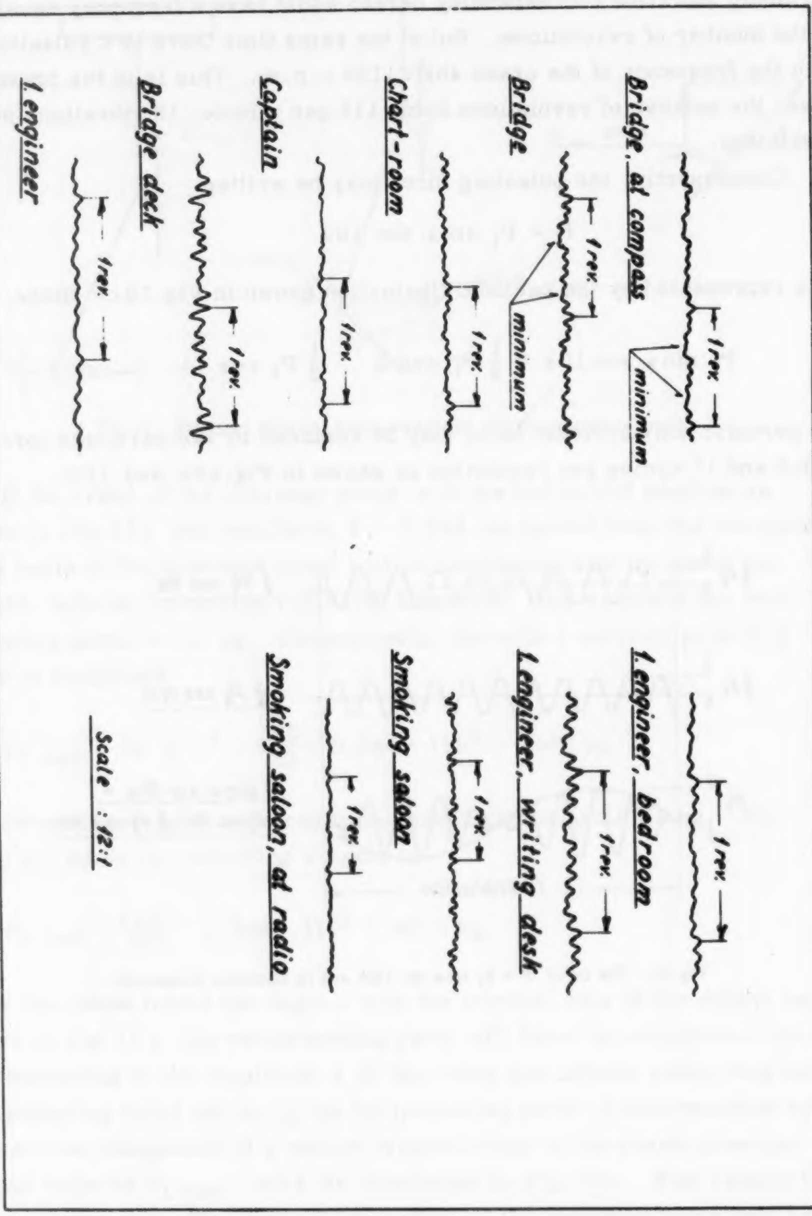

Fig. 20. Vibrograms for ship N° 3. Vertical vibrations, 115 r.p.m.

Fig. 20 shows vibrograms taken at various positions on the ship where the vibrations were greatest. It is seen that the vibrations are very like the vibrations shown in Fig. 19. This makes it rather clear that it is really the torsional vibrations in the crank shaft that produce the vertical vibrations in the hull.

The impulse forces for the cylinders are calculated in the same manner as for the scavenge pump and are stated in the table below.

Cylinder	Amplitude on crank radius	Reciprocating weights	Revolving weights	$P_{t\ osc}$	$P_{t\ rot}$
	cm	kg	kg	kg	kg
Sc. p.	0.244	555	1165	1960	4110
1	0.314	5700	3970	25900	18000
2	0.268	5700	3970	22100	15400
3	0.190	6000	3970	16500	10900
4	0.033	5700	3970	2720	1900
5	-0.077	6000	3970	- 6690	- 4430
6	-0.177	5700	3970	-14600	-10200
7	-0.255	6000	3970	-22100	-14600

Fig. 21 shows the crank positions and the corresponding directions of crank displacement for the cylinders and the scavenge pump. On the base of this scheme and of the pulsating forces stated in the table page 30 the vector diagrams for the reciprocating and revolving parts are drawn as shown in Fig. 21. The resulting forces $\sum P_{t\ osc} = 1650$ kg and $\sum P_{t\ rot} = 3050$ kg are added vectorially into $\sum P_t = 4010$ kg. This vector is represented by the dotted line in Fig. 19. The periodic force may therefore be written

$$P = 4010 \sin\alpha \sin 10\alpha \text{ kg}$$

The horizontal free force due to the revolving parts has the value 3050 kg.

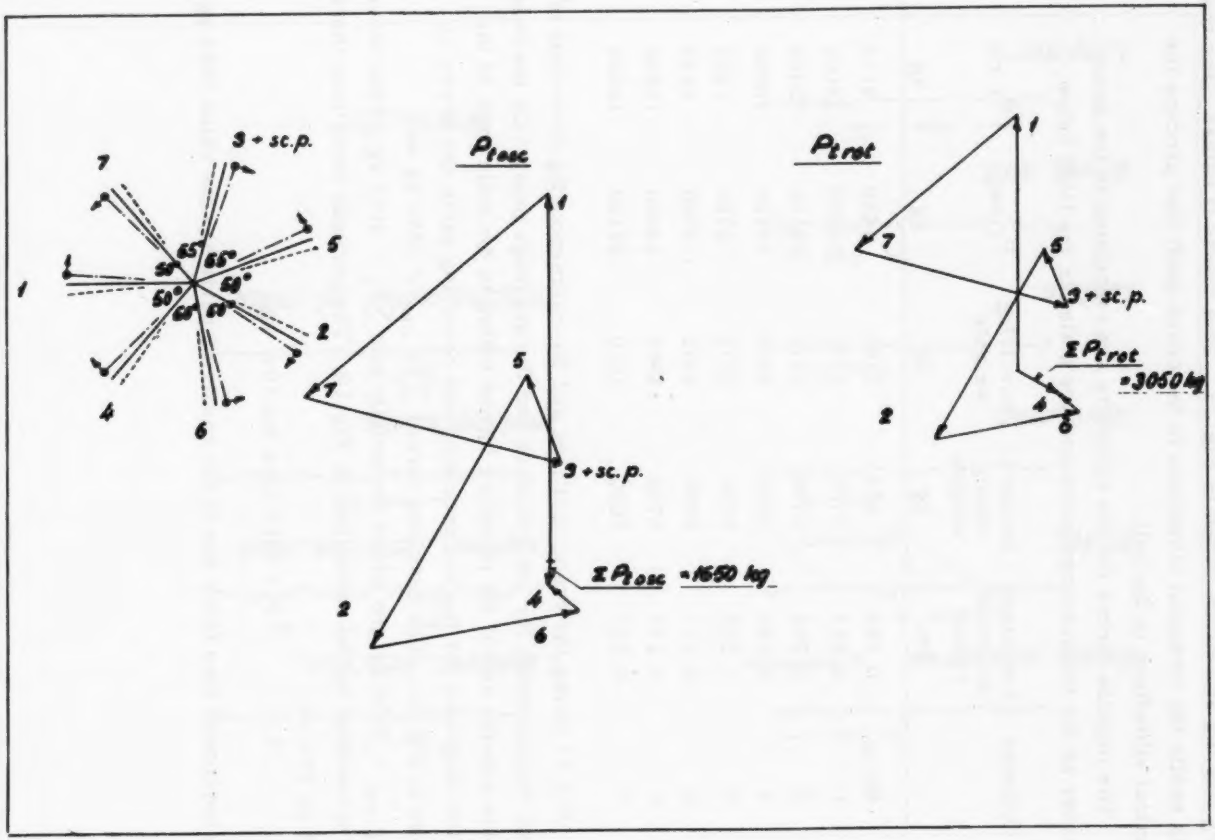

Fig. 21. Force diagrams for reciprocating ($P_{t\ osc}$) and revolving ($P_{t\ rot}$) parts.

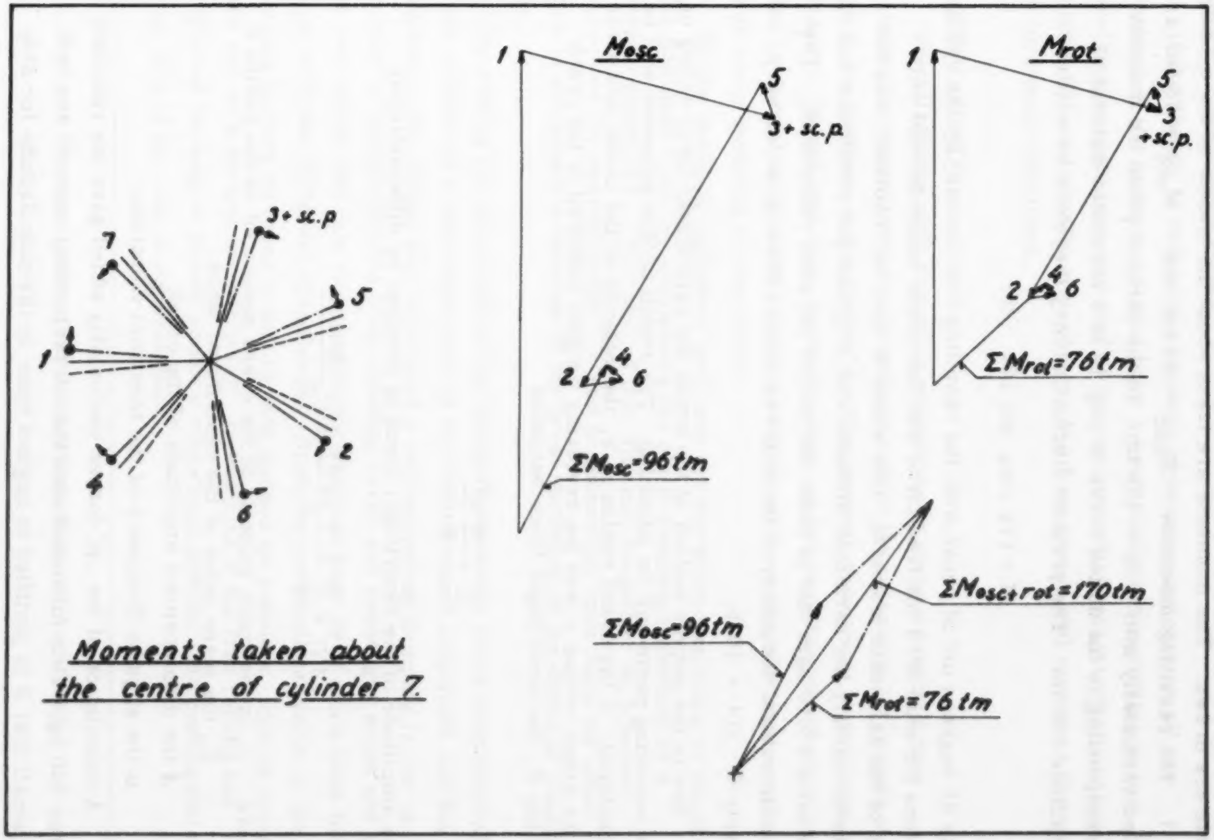

Fig. 22. Moment diagrams for reciprocating (M_{osc}) and revolving (M_{rot}) parts.

Besides the free forces also free moments are acting. In Fig. 22 the vector diagrams for the moments of the reciprocating and the revolving parts are drawn. The moments are taken about the middle of the cylinder Nº 7. The resulting moments M_{osc} = 96 tm and M_{rot} = 76 tm are added vectorially into M = 170 tm. In the vertical plane the moment corresponding to the dotted curve in Fig. 19 is a harmonic moment of amplitude 170 tm. The periodic fluctuation may therefore be written

$$M = 170 \sin\alpha \ \sin 10\alpha \quad \text{tm}$$

It is 90 degrees out of phase with the resulting free force. By the calculations for ship Nº 3 the free force was therefore neglected and only one of the harmonic moments, into which M may be resolved, was considered namely the harmonic moment with 9 cycles per revolution and of the value 87000 kgm that is about the half of the peak value of M. The 11-node natural frequency of the hull was about 1035 c.p.m. or very nearly $9 \cdot 114 = 1026$.

Due to the uneven motion of the crank the centrifugal force acting on the revolving parts will be pulsating. The result of this pulsation will be considered. If the crank radius is r, the velocity of the crank measured at the crank radius v and the revolving weights referred to the crank radius W, the centrifugal force becomes

$$C = \frac{W}{g}\frac{v^2}{r}$$

The amplitude of the centrifugal force is obtained by differentiating

$$(\Delta C) = \frac{W}{g}\frac{2\ v\ (\Delta v)}{r} = C_o \frac{2\ \Delta\ \omega}{v_o}$$

where v_o is the mean velocity of the crank measured on the radius r
C_o the mean value of the centrifugal force
Δ the displacement amplitude of the crank
ω the angular frequency of the torsional vibrations.

A calculation of the ΔC-forces vectorially added gave the resultant force 600 kg. This force and also the corresponding moment are both so small that it is justified to neglect them in the calculations for ship Nº 3.

Dismounting the flywheel it should be possible to obtain a sensible increase of the natural frequency of the crank shaft and at the same time

a sensible reduction of the torsional vibrations. This was done and in confirmation of the theory it resulted in an appreciable reduction of the vertical vibrations in the hull.

If the engine is balanced by means of balance weights on all the cranks and the balance weights have a magnitude corresponding to the revolving weights plus one half of the reciprocating weights these balanced weights will practically also balance the free forces and moments due to the torsional vibrations.

8. Final remarks.

Investigations with regard to vibrations in ships were until now principally concentrated on the calculation of the natural hull frequencies and of the position of the nodes *). The general object was to avoid resonance. Seeing that this, in most cases, is not possible, the Author has tried to solve the problem of calculating in advance the vibration amplitudes corresponding to resonance.

By the methods published until now the vibration amplitude at resonance has been found by multiplying the static bending deflection by a dynamic magnifier. The value of this dynamic magnifier is, however, only stated for special cases **) and to generalize upon them for other ships is not possible because of the great variation of the dynamic magnifier.

By putting the amplitude of the damping harmonic force proportional to the product of a waterline area by the displacement amplitude and the frequency, the Author has calculated a corresponding damping factor on the base of measurements from existing cargo and passenger ships and it has been found that this damping factor does not vary very much even for ships of rather different dimensions. Applying the smallest value of the damping factors so found it will by this method be possible, without much work and with an appropriate safety, to examine if the free forces and couples of the engine should produce vibrations greater than permissible in the hull of the ship being designed.

---

*) J.L. Taylor: Transactions of the Institution of Naval Architects (I.N.A.) 1930

C.W. Prohaska: Bulletin Association Technique Maritime 1947

A.J. Johnson: B.S.R.A. Report No. 19, July 1948

**) A.J. Johnson: Transactions North East Coast Institution 1951

It may be supposed that a continuation of the investigations will enable more accurate values for the constants to be obtained thereby improving the calculation of the vibration amplitudes.

The suggested theory has, furthermore, in one case been applied to the propeller excited vibrations with frequency: number of blades multiplied by the number of engine revolutions, and in another case to the vibrations due to the torsional vibrations in the crank shaft. Even if these calculations are connected with some uncertainty they seem to confirm that the method may be applied to high frequency vibrations with the same value of the damping factor as for lower frequencies.

Acknowledgement:

In the preparation of this paper the author is primarily indebted to professor H. Nilou for his invaluable help in checking testresults and for making many helpful suggestions concerning the text as well as for the translation into english.

LAST ISSUES OF ACTA POLYTECHNICA

Mechanical Engineering Series

(the predecessor of Acta Polytechnica Scandinavica)

Volume 2

Nr 1 ZACHRISSON, L E: *On the Membrane Analogy of Torsion and its use in a simple apparatus.* Acta P 80 (1951), 40 pp, Sw. Kr 4: 00 UDC 539.414.08

Nr 2 LINDBLAD, A: *Some Experiments with Self-Propelled Models of Twin Screw Ships.* Acta P 89 (1951), 24 pp, Sw. Kr 4: 50 UDC 629.12.001.5:532.385.5

Nr 3 PIERRE, B: *Influence of Frames on Insulation of Cold Storage Chambers on Board Ship.* Acta P 92 (1951), 65 pp, Sw. Kr 7: 00 UDC 629.12.011.516

Nr 4 LUNDBERG, G, and PALMGREN, A: *Dynamic Capacity of Roller Bearings.* Acta P 96 (1951), 32 pp Sw. Kr 5: 00 UDC 539.313 621.882.7.8

Nr 5 KARLSON, K G: *Sur le frein à sabots extérieurs articulés.* Acta P 100 (1951), 31 pp, Sw. Kr 5: 00 UDC 621-592.112

Nr 6 WIDELL, T A, and JUHASZ, S I: *Metal Temperature in Regenerative and Recuperative Air Preheaters.* Acta P 103 (1952), 50 pp, Sw. Kr 6: 50 UDC 625.925:620.19

Nr 7 PETERSSON, S: *Investigation of Stress Waves in Cylindrical Steel Bars by Means of Wire Strain Gauges.* Acta P 118 (1953), 22 pp, Sw. Kr 3: 50 UDC 534.1.08

Nr 8 FALKEMO, C: *On the Possibilities of Estimating the Towing Resistance of Ships by Tests with Small Models. I.* Acta P 122 (1953), 51 pp, Sw. Kr 7: 00 UDC 629.12.072.5.001.57

Nr 9 ODQVIST, F: *Influence of Primary Creep on Stresses in Structural Parts.* Acta P. 125 (1953), 18 pp, Sw. Kr 2: 50 UDC 539.434

Nr 10 OLSSON, U: *Non-circular Cylindrical Gears.* Acta P 135 (1953), 214 pp, Sw. Kr. 18: 00 UDC 621.833.5.621.834

Volume 3

Nr 1 PETTERSSON, O: *Circular Plates Subjected to Radially Symmetrical Transverse Load Combined with Uniform Compression or Tension in the Plane of the Plate.* Acta P 138 (1954), 30 pp, Sw. Kr 7: 00 UDC 539.384.4

Nr 2 NIORDSON, F: *Vibrations of a Cylindrical Tube Containing Flowing Fluid.* Acta P 139 (1954), 28 pp, Sw. Kr 4: 00 UDC 534.131.2

Nr 3 NIORDSON, F: *Vibrations of Turbine Blades with Loose Hinge Support.* Acta P 140 (1954), 11 pp, Sw. Kr 5: 00 UDC 534.014.1:621-253.5

Nr 4 KARLSSON, K I: *Simple Calculation of Deformation and Stress in the Shell of Thin-walled Cylindrical Vessels.* Acta P 162 (1954), 24 pp, Sw. Kr 8: 00 UDC 539.384.6

Nr 5 STENEROTH, E: *On the Transverse Strength of Tankers.* Acta P 169 (1955), 104 pp, Sw. Kr 11: 00 UDC 539.4:629.123.56

Nr 6 DAVIDSSON, W: *Investigation and Calculation of the Remaining Tensile Strength in Wire Ropes with Broken Wires.* Acta P 174 (1955), 38 pp, Sw. Kr 8: 00 UDC 621.86.065.3:620.172

Nr 7 GÖTZLINGER, J, and JOHNSSON, S: *Dynamic Forces in Cranes.* Acta P 175 (1955), 34 pp, Sw. Kr 8: 00 UDC 621.873.001.2:621.873:531.78

Nr 8 SCHRØDER, H: *Distribution of the Surface Pressure over the Friction Lining of Brakes with Rigid Shoes and Drums, and Efficiency of Such Brakes.* Acta P 188 (1956), 44 pp, Sw. Kr 6: 00 UDC 621-592:629.113

Nr 9 FALKEMO, C: *On the Possibilities of Estimating the Towing Resistance of Ships by Tests with Small Models, II* Acta P 201 (1956), 15 pp. Sw. Kr 5: 00 UDC 629.12.072.5.001.57

Nr 10 KARLSSON, I: *On the Stability of the Steel Lining in a Vertical Pennstock Shaft* Acta P 211 (1957), 22 pp Sw. Kr 8: 00 UDC 627.8.014.2:627.8.044

Nr 11 TOFTDAHL OLESEN, H: *Calculations on Jog Conveyors.* Acta P 220 (1957), 20 pp. Sw. Kr 5: 00 UDC 621.867.52

ACTA POLYTECHNICA SCANDINAVICA

Mechanical Engineering Series

Me 1 COLDING, B: *Testing of Machinability by Radioactive Methods.* (Acta P 243/1958) 42 pp. + figs. Sw. Kr 7: 00 UDC 621.9.025:620.178.16:539.16.004.14

Me 2 KJÆR, VIGGO A, *Vertical Vibrations in Cargo and Passenger Ships.* (Acta P 244/1958) 52 pp. Sw. Kr 7:00 UDC 534.838.1:629.12.07

Price Sw. Kr. 7,00

ATOR TRYK

YK